A DYNAMIC DAMPING DEVICE FOR PAYLOAD PENDULATIONS OF CONSTRUCTION CRANES

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(ABSTRACT)

As a material handler, the crane plays a vital role within the operations of the manufacturing, construction, and shipping industries. Objects of all shapes and sizes are conveyed with the crane to improve productivity and reduce worker fatigue. The crane's capacity to operate efficiently and safely however, suffers from payload pendulations. This cyclic motion of crane cable and payload produces schedule delays, property damage, and high risk to personnel.

Current pendulation reduction systems have typically been applied to overhead cranes within the manufacturing and shipping industries. The construction industry in contrast, has failed to innovate tower and mobile cranes. This can be traced to the complexity of the construction operation and the conditions under which the construction crane performs.

This thesis aims to improve productivity and safety within the construction industry through the application of damping systems on construction cranes. To achieve this goal, an experimental model will be developed and tested. The design process will include an analysis of operational constraints, theoretical design, and physical testing.

Tuned mass damping will be investigated as the basis for the damping control method. Theory will be detailed and incorporated in a mathematical simulation. The tuned mass damper, a cantilevered rod, will be designed and tested for application. The system will then be coupled to a scaled crane model for testing. Data analysis will be used to define the models effectiveness. From the theoretical analysis and physical testing, a conceptual model will be defined. Subjects for future research will also be presented.

ACKNOWLEDGMENTS

The author would like to thank his parents and family for providing encouragement throughout his life. Without their support, this thesis would never have been written.

The author would also like to thank his thesis committee for providing guidance and valuable knowledge throughout the research process. Faculty, staff, and fellow students are likewise thanked.

TABLE OF CONTENTS

LIST OF FIGURES, GRAPHS, AND TABLES

CHAPTER 1

INTRODUCTION

The struggle to achieve efficient material handling can be found in even the earliest civilizations. The first cranes were designed to enhance the productivity of the human worker. With this breakthrough, larger and more complex objects could be lifted and placed. Further control advancements improved the speed and productivity of repetitive lifts. Later breakthroughs led to the creation of tower, mobile, container, and overhead cranes for completion of specialized tasks. In combination, these technologies generated an atmosphere in which the manufacturing, construction, and shipping industries could flourish. Today, the crane has become a critical component of these operations.

The dramatic development and utilization of the crane has however, been plagued by what seemed to be an unavoidable problem. Despite the creation of larger and more complex designs, payload pendulations limit productivity. This phenomenon is exhibited by the uncontrolled oscillation of payloads during the conveyance cycle. Problem sources are attributed to wind, operator error, poor crane design, and a number of other factors. This produces operation delays and unpredictable costs. In addition, safety is compromised when attempting to perform procedures under conditions the crane was not intended.

Selected industries have worked to overcome payload pendulations. The re-engineering of overhead cranes through technological advancement provides an excellent example. Manufacturing and shipping industries required accurate and efficient transport of loads within confined spaces. Oscillation of loads at the beginning and end of the movement cycle however, restricted the pace of production while compromising safety. This was later linked to variance in conveyance speed caused by operator error and shortcomings in crane design. The incorporation of damping technologies, like computer optimization of the crane control path, have dramatically improved efficiency. Ultimately, this damping system has reduced production times and increased the safety of these operations (Hubble, et al., 1992; Ridout, 1989; Virkkunen, et al., 1990).

Unfortunately, technological systems designed to damp payload pendulations are not commonly implemented on construction sites. Construction depends upon cranes for the accurate placement of inventory in almost every phase of production. In many cases, operations cannot be performed without cranes. This creates a critical environment in which payload pendulations can produce delays. Methods such as operator control and tag lines can be used to control pendulations in some circumstances. However, when weather conditions are poor or inexperienced crane operators are used, the crane's productivity suffers. Delays result and money is lost to accidents, damaged inventory, and compromised schedules.

Why have innovative attempts to control payload pendulations on tower and mobile cranes not been pursued? The failure to advance these systems can be credited to both the demands of the operation and the environment under which the crane functions. Construction occurs under many conditions. The type of loads to be moved, the height, accuracy, position, and distance of placement as well as the available space of the site and a number of other factors change continuously. In addition, operations are performed under a wide range of ground and weather conditions. These characteristics, individually or in combination, create an unpredictable atmosphere requiring a design capable of operating "well" under many conditions. Most commercially available damping systems are unable to meet the constraints of this dynamic operation.

A closer examination of overhead cranes reveals the design and operational attributes that have allowed the control of payload pendulations. Unlike tower and mobile cranes, overhead and gantry cranes have a more predictable conveyance path. Since the rigid frame limits the placement area, the designer can predict the beginning and end of the movement cycle with some accuracy. This allows for optimization of the control. Typically motor speed is increased or decreased according to site position in an effort to reduce the cable angle during movement (Jones, et al., 1988; Ridout, 1989; Sagara, et al., 1990; Virkkunen, et al., 1990).

The performance and predictability of overhead cranes is further enhanced by their design constraints. The rigid structure of the frame removes pendulation sources like ground movement beneath the crane and boom deformation under load. In addition, these cranes are typically operated within enclosures which limit the effects of wind upon the operation. Together these changes produce a system with few pendulation sources. The remaining sources are directly linked to operator control during movement.

1.1 Motivation

It is believed that the incorporation of pendulation damping systems on mobile and tower cranes would be instrumental in increasing productivity and safety within the construction industry. The application of damping systems on industrial overhead cranes has suggested the ability to improve efficiency. Similar systems could be used to improve safety and increase productivity in the construction industry. In addition, the reduction of damaged inventory combined with an ability to work in a wider range of weather and site conditions would reduce schedule delays. Current damping systems (the use of tag lines and operator damping control) could also be reduced thus freeing these workers to concentrate on the task at hand.

The possibility of increasing productivity alone is motivation to investigate the subject of dynamic damping systems. Construction as a business relies upon the scheduling of independent activities which have been critically linked. Since the industry depends upon efficient crane material handling, it is imperative that the crane be capable of functioning properly under the majority of conditions. Should an activity be delayed due to reduced crane performance, following activities and the project as a whole are subject to delay. This leads to considerable cost to both the owner and contractor.

Some construction activities require the completion of repetitive material lifts with cranes. The conveyance of concrete using two cubic yard (2 CY) buckets and the placement of steel support structures are good examples. These operations are defined by multiple lifts to complete a single operation. Payload pendulations create inefficiency during each lift and place cycle. Crane damping systems could reduce the time required for each cycle and eventually lead to large gains for the entire operation. The reduction of actual construction time and damage to inventory could drastically reduce the cost of an industry responsible for ±10% of the Gross National Product. Likewise, the skill level required for crane operators could be reduced.

Safety is another critical factor. The construction industry employs 5% of the workforce, but generates 20% of all accidents and 26% of all occupational related fatalities (Della-Giustina, 1991). In a study conducted in 1978 in the United Kingdom, lifting equipment accounted for 9% of construction fatalities and 1.5% of injuries (Helander, 1991). A similar study conducted between 1977 and 1989 in Finland found that lifting equipment accounted for 4.8% of construction accidents (Niskanen, et al., 1988). The striking of laborers by swinging payloads accounts for 39% of crane accidents (Jarasunas, 1990). Crane damping equipment by reducing the frequency and amplitude of payload oscillations could help in preventing these injuries.

The injury of workers is not only tragic but very expensive. Direct costs include rehabilitation, workman's compensation, hospital stays, and lost work time. Indirect costs are associated with construction insurance, operation profitability, schedule delays, and a number of other factors. It has been estimated that indirect costs resulting from construction accidents are over 4 times greater than direct costs (Heinrich, 1959). All of these costs ultimately effect the success or failure of the project. The direct cost of construction accidents amount to roughly 6% of building costs (Helander, 1991). These in addition to the even greater amount of indirect costs are later passed on to the consumer.

1.2 Objectives

This thesis seeks to develop a conceptual model of a passive damping system for construction cranes. This objective will be achieved through a series of steps. First, an investigation and initial computer simulation of tuned mass damping theory will be performed. From this foundation, an experiment will be conducted to confirm the physical properties of cantilevered beams for later inclusion as damping devices. A second experiment will then be conducted to explore the effects of tuned mass damping on a scaled crane cable model. This will suggest the effectiveness of such a system in industrial application. With the experiments concluded, an analysis of expected and achieved damping results can be made. This will define special considerations to be made in the conceptual crane damping system.

Ultimately the research aims to contribute to the construction industry through investigation of damping technologies. The conceptual model could be explored in future research to produce a working system. The intention is to increase crane productivity on repetitive lift activities while improving overall job safety.

1.3 Scope and Limitations

This thesis will investigate the application of cantilevered beams operating as tuned mass damping systems to reduce payload pendulations experienced on mobile cranes. The research will produce a conceptual model of an industrial crane damping system for future research and production.

The scope of this research will be limited to the analysis and application of cantilevered beams in tuned mass damping systems for cranes. Discussion of current construction practice and pertinent damping theories will be provided as a baseline from which to establish the validity of this work. Some exploration of other areas of damping will be incorporated as a format to suggest the positive attributes of such a system.

1.4 Methodology

A series of tasks will be performed in the research process. These will involve both literature review and physical experimentation. The results will be incorporated into the thesis as narrative material and quantifiable data.

1. The research will begin with an investigation of crane payload pendulations. This will lead to a baseline understanding of the subject and potential for improvement within the construction industry.
2. The theory from which tuned mass damping systems are developed will be explored. Computer simulation using the program Matlab will then be conducted to provide a foundation for further research.

3. Cantilevered beams will be researched as the tuned mass damping method. This will involve the development of formulas for computer simulation and physical testing.
4. Physical testing will be conducted in a series of two experiments. The first will generate data to confirm the predicted properties of cantilevered beams. The second will incorporate the coupling of a cantilevered rod and scaled crane cable model. This will allow physical testing of the cantilevered rod as a tuned mass damper.
5. A conceptual model of a crane passive damping system will be developed. The design will be based upon the analysis of data produced in the completion of the computer simulation and the two physical models.

1.5 Presentation of Thesis

This thesis presents the results of an investigation of tuned mass damping theory. The theory will be incorporated in the creation of a conceptual damping system for industrial application on construction cranes. An emphasis will be placed on maintaining the crane's original functionality while improving it's productivity. Areas for exploration with the intention of advancing material handling in construction will be suggested in the final design.

Chapter 2 - Tuned Mass Damping Theory

An investigation of tuned mass damping theory will be presented. The chapter will begin with a preliminary discussion to familiarize the reader with the basic concepts which define tuned mass damping. This will be followed by a presentation of the basic design in the form of a simple pendulum model. The chapter will then expand the theory to include cantilevered beams as a damping source. Other chapter topics include a comparison of passive and active damping as well as the constraints under which a construction crane system must be designed.

Chapter 3 - Cantilevered Rod Theory

Chapter 3 will build upon the ideas presented in chapter 2. This will begin with an examination of the cantilevered rod as a damping system. The presentation of frequency equations and an equation for the critical buckling load will then follow. This is considered vital to the thesis as it defines the rod's properties for later investigation in the physical experiments. The section will also suggest design considerations for the application of a damping rod. Finally, a simulation of the cantilevered damping rod will be presented.

Chapter 4 - Cantilevered Rod Experiment

This chapter will present the creation and testing of a cantilevered beam model. Upon evaluating the properties of the material and size of cantilevered rod chosen, a scale model will be constructed. This model will be designed for testing performed to confirm the expected character of cantilevered beams as suggested by the formulas provided in chapter 3. The measurement of the rod's period of oscillation under various conditions will be a primary consideration. Construction, testing, and findings will be detailed. Conclusions pertinent to the conceptual crane damping system will be discussed.

Chapter 5 - Tuned Cantilevered Rod

To understand the effects of tuned mass damping and the application of cantilevered beams as tuned mass dampers, a second experiment will be presented in chapter 5. This will include the coupling of the cantilevered rod designed in chapter 4 to a scaled crane cable model. From this model a series of experiments will be conducted to evaluate the use of cantilevered rods as damping systems. The chapter will present quantifiable data and narrative explanation to establish the principles defining the conceptual model.

Chapter 6 - Conceptual Crane Model

Upon the completion of the physical models, the research will focus upon the creation of a conceptual crane damping system. This model will take into account all factors leading to the damper choice and system function. A discussion of the measuring systems, mechanical systems, and support systems will be provided. The main goal of this section will be to suggest the expansion of the research to future control theory and other areas of interest.

Chapter 7 - Summary and Conclusions

Thesis conclusions will be presented and evaluated. A review of the research conducted and the significance of the work will also be provided.

CHAPTER 2

TUNED MASS DAMPING THEORY

As mentioned in the introduction, this thesis is intended to develop a conceptual model of a dynamic damping device to be incorporated on construction cranes. In this respect some consideration was given to the type of device to be created. This would include an examination of the damping system's operation, size, and resistance to failure.

When considering damping devices for application, a choice must be made between passive damping systems and active damping systems. Passive damping systems operate by allowing a transfer of kinetic energy from the vibrating or oscillating object to the damper. The energy is then controlled and dissipated through the devices internal damping. This mode of operation requires little design complexity. With few moving parts, the risk of failure due to malfunction is low. Active damping systems in contrast, work by imparting a catalyst which would not have occurred naturally. This requires greater complexity and risk. Should the active damper malfunction, the catalyst could cause a detrimental increase in the oscillation or vibration.

As a potential drawback, purely passive damping has not proven to be as efficient as active damping in its intended use. Active damping is created by taking a natural reaction and enhancing its operation with an applied force or movement. Because the force is man-made, the amplitude and time of application can be controlled for best results. While passive damping can not amplify a natural reaction, some control of the reaction can be implemented to achieve similar results. This changes the passive damper to what can be considered a semi-passive state. The system can now be used for a number of conditions with slightly greater complexity.

To know when semi-passive or active damping should be used, the designer must consider more than just the advantages and shortcomings of its operation. For instance, the environment of its application plays a large role in the design's success. If the damper is to be employed in hostile weather such as wind, rain, mud, or snow, the use of complex systems of moving parts would not be encouraged. Application on cranes provides an excellent example. Construction occurs during every weather condition in every environment of the world. A damping device operating on a construction crane must therefore be reasonably efficient under a wide range of conditions. It is clearly advantageous to reduce the number of integrated parts if a design is to be generated for a robust damper that will function well without failure.

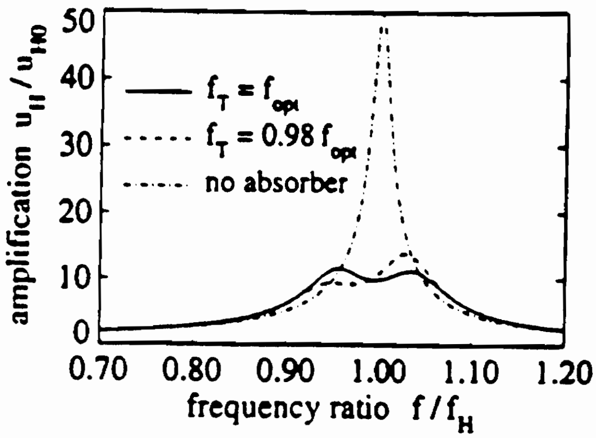
A second design consideration stems from the predictability of the vibration source. Oscillations can be caused by a number of effects. If a machine has only one mode of operation which occurs in only one location, the problem source can easily be identified. Equipment which is employed over a large number of uses and terrain, can incur vibration from numerous reactions. In this respect, the operation of mobile construction cranes can be considered very complex. Crane payload oscillation is attributed to wind, poor crane design, operator error, ground movement beneath the crane, and a number of other sources and combinations. Because it is next to impossible to predict the problem source, a damping system designed for a few conditions will not be efficient or even functional. For example, active damping systems which typically operate by predicting a vibration and then producing a preventative reaction would be limited in their success. In contrast, semi-passive damping systems typically absorb or control vibration without regard to its source. This improves the range of application and makes semi-passive dampers suitable for utilization on construction cranes.

A review of the arguments for and against different damping systems, provides a better understanding for potential application on construction cranes. The wide range of operations and environments under which cranes operate is a critical factor. Damping systems must possess the ability to function over a number of weather and site conditions. In addition, the system must be robust and efficient for controlling vibration from a number of sources. Semi-passive damping seems to meet these constraints. In particular, tuned mass damping would be optimum under the various conditions.

2.1 Preliminary Discussion Of Tuned Mass Damping

Tuned mass damping, like most forms of damping, is intended to reduce the kinetic energy (KE) of a system thus bringing it to rest. The kinetic energy of the object is typically in the form of motion occurring as vibration or oscillation. In the case of crane operation, the cable and payload act as a pendulum. The movement of the crane or gusts of wind increase the pendulum's potential energy (PE) by displacing it radially from its lowest position. From this point the principle of conservation of mechanical energy takes effect. The displaced payload attempts to reduce its potential energy by swinging to the lowest achievable position. Unfortunately, the pendulum's momentum continues its movement to another position of potential energy at the opposite side. The effect is a cyclic oscillation in the form of PE - KE - PE. This motion if graphed would resemble a ~~sign~~ sine wave with a measurable frequency, *f*, defined as the number of complete oscillations per a second. It is this uncontrolled movement that reduces the crane's productivity and operation safety.

Two basic principles are associated with the operation of tuned mass dampers. The first is defined by a frequency change. This is typically examined with the use of amplification curves as seen in Figure 2.1.



(Bachmann, et al., 1995)

Figure 2.1 Amplification Curves

The vertical axis corresponds to a ratio between the displacement of an oscillating structure, U_H, and the structure's quasi-static displacement, $U_{HO} = \Delta G_i / k_h$. ΔG_i is the force amplitude of the ith harmonic of the excitation force and k_h is the structural spring constant. The horizontal axis represents a ratio between the frequency of excitation, f, and the structure's eigenfrequency or characteristic natural frequency, $f_H = \frac{\sqrt{\frac{k_h}{m_h}}}{2\pi}$ in which m_h is the structure's mass. The plots represent a structure's reaction to a sinusoidaly applied excitation force in terms of oscillation amplification at specific frequencies. The various plots correspond to the structure in an undamped mode (no absorber), an optimized damper of eigenfrequency, $f_T = f_{opt} = \frac{\sqrt{\frac{k_T}{m_T}}}{2\pi}$, and a non-optimized damper with frequency $f_T = .98 f_{opt}$.

In figure 2.1 a single large amplification peak represents the undamped periodic oscillations of a structure. This is commonly modeled as a single degree of freedom system. According to the theories developed by Frahm in 1911 (Bachmann, et al., 1995; Smith, 1988) the addition of a tuned frequency splitter changes the amplification curve by creating a new structural system with new frequencies. The energy residing under the single peak is now redistributed to produce two peaks of lower amplification. One peak will occur at a frequency above the eigenfrequency of the original structure and the other will be below the eigenfrequency of the damper alone. This is represented in Figure 2.1 as $f_t = .98 f_{opt}$ where an absorber with non-optimized frequency is attached.

The second function of the absorber is to control and dissipate the structure's energy. In 1928 Ormondroyd and Den Hartog (Smith, 1988) improved the application of the tuned frequency splitter by adding a damping mechanism. The tuned frequency splitter can now impart an effect over a greater range of frequencies. In addition the mechanism can absorb the structures energy, control it, and then dissipate it through internal damping.

Hahnkamm in 1933 (Smith, 1988) suggested that the most efficient absorber is achieved when the amplitudes of the higher and lower frequency peaks are equivalent. This is represented in Figure 2.1 as $f_t = f_{opt}$ where an absorber of optimized frequency is attached. To achieve this the absorber's weight relative to that of the structure is taken into consideration. According to this theory, choosing a larger or smaller damping mass effects the natural frequency required to achieve optimum performance. Typically an absorber to structure mass ratio, μ, of 1 to 2% is used (Bachmann, 1992). The vibration absorber can now be considered a "tuned mass damper". This condition is satisfied by the ratio

Equation 1 $$\frac{AbsorberFrequency}{MainSystemFrequency} = \frac{1}{1+\mu}$$ Frequency Optimization Ratio

Further research by Brock in 1946 (Smith, 1988) revealed that optimum tuned mass damping is achieved when the system creates the flattest frequency peaks possible. This is achieved by designing for a ratio of optimum structural damping, ζ_s. An average value often used for design is defined by

Equation 2 $$\zeta_s = \sqrt{\frac{3\mu}{8(1+\mu)^3}}$$ Optimized Structural Damping

In physical terms ζ_s could be likened to the rate at which floor vibrations or pendulum oscillations are brought to rest.

It is important to note that the oscillation of the tuned mass damper is much greater in amplitude than the structure to be damped. Here Bachmann et al. suggest that greater mass ratios yield lower absorber amplitudes (Bachmann, et al., 1995). This has an effect upon the design's application. The damper should be chosen to reflect the fatigue and space required to achieve the desired effect. The relative motion of the absorber mass was derived by Den Hartog in 1947 (Smith, 1988).

2.2 Simple Pendulum Model

The implications of the previously discussed theories can be suggested in an example. A series of coupled simple pendulums as shown in Figure 2.2 provides a good model. The first pendulum, Pendulum 1, is oscillating uncontrollably. The damping pendulum, Pendulum 2, is meant to bring Pendulum 1's oscillations to rest by absorbing its energy and dissipating it through internal damping.

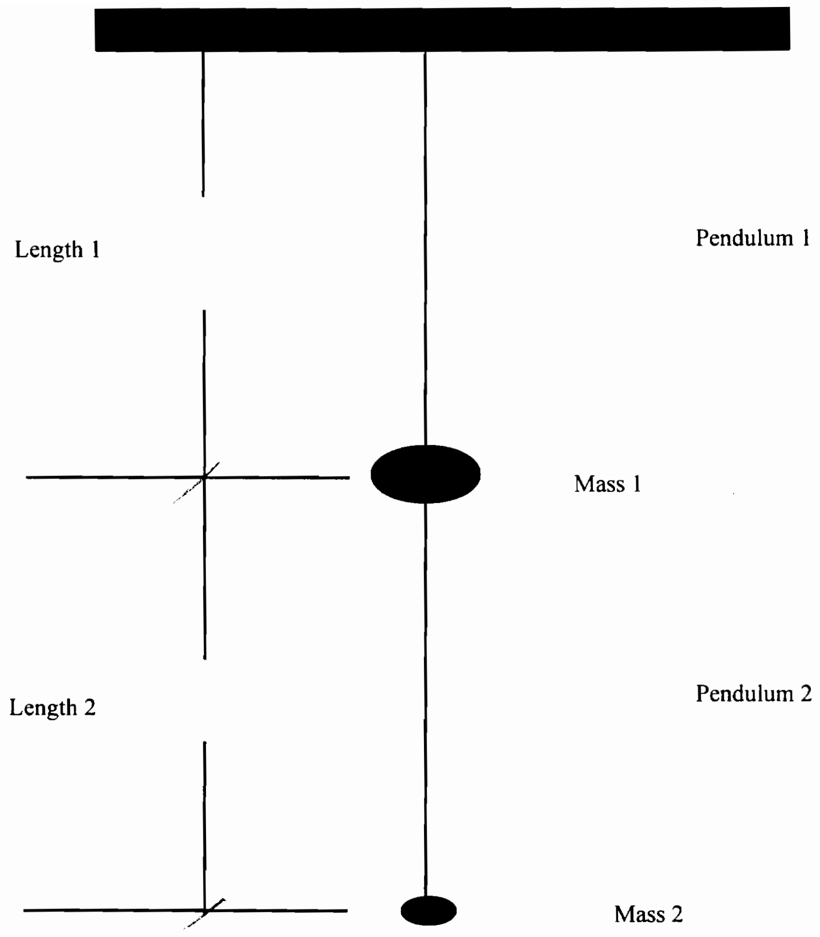


Figure 2.2 Simple Pendulum Damper

A simple pendulum oscillates at a frequency defined by the equation,

Equation 3 $$f=\frac{1}{2\pi}\sqrt{\frac{G}{L}}$$ Simple Pendulum Frequency

where G is equal to gravitational acceleration and L is equal to cable length.

From this equation it can be found that the length of the cable determines the pendulum's period. Having chosen an appropriate damping mass, Mass 2, the damping pendulum should be tuned to the appropriate frequency to achieve the desired damping. Taking into consideration the work of Frahm, Ormondroyd, Den Hartog, and Hahnkamm, the length of Pendulum 2's cable is adjusted to achieve the optimum natural frequency relative to that of the main pendulum.

As an example, if Pendulum 1's length was 100 Meters (M) and Mass 1 was 300 Kilograms (Kg) the frequency as calculated from Equation 3 would be approximately .05 Hz. If Mass 2 was chosen to be 10 Kg or 10% of Mass 1, Equation 1 could be used to find the optimized absorber frequency, approximately .045 Hz.

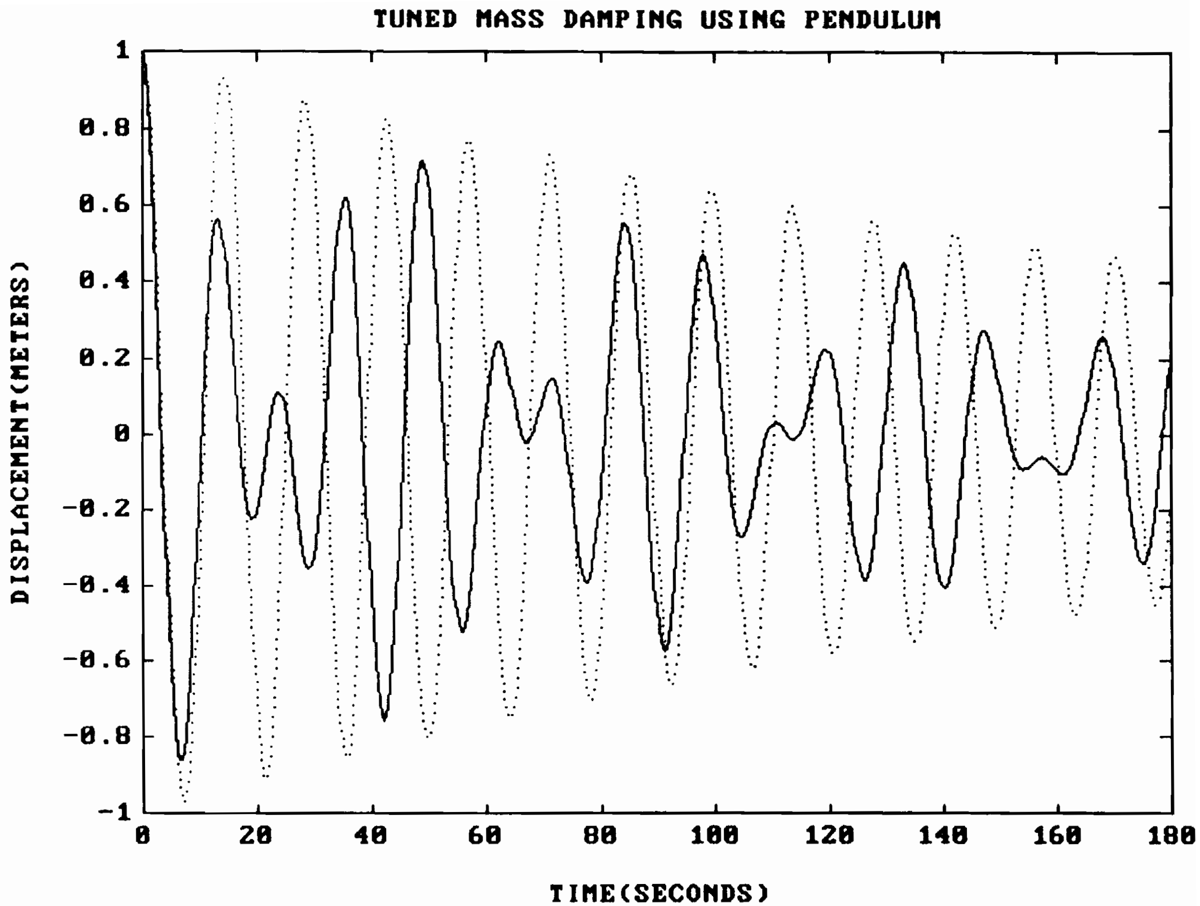
$$\frac{AbsorberFrequency}{.05Hz} = \frac{1}{1+.1}$$

Using Equation 3 once again, the cable length required by Pendulum 2 to achieve .045 Hz would be approximately 123 M.

In operation the system creates a damping effect. A catalyst force, such as wind, displaces Pendulum 1 and 2 to a position of higher potential energy. The coupled system then enters a form of cyclic oscillation. However, the initial energy of Pendulum 1 is transferred to Pendulum 2. In effect, Pendulum 1's oscillations are reduced and Pendulum 2's oscillations are amplified.

As a test of the tuned mass damping theory, a series of computer simulations were conducted on the program Matlab. The program, included in Appendix A, incorporated a time study of the motion of two coupled pendulums during oscillation. The coupled system's reaction to release from an arbitrary initial deflection can be observed in Graph 2.1. This is a depiction of two coupled pendulums with 50 meter cable lengths and mass of 3628 Kg and 363 Kg respectively for Mass 1 and Mass 2.

The Matlab simulation is significant for several reasons. First, it can be noticed that the damped pendulum's magnitude of oscillation was significantly reduced in 2 cycles or approximately 30 seconds. A similar pendulum oscillating without the coupled damper would have swung for a much greater time period. The only pendulation reduction would have occurred from resistance provided by wind, crane motion, the cable itself, and several other insignificant factors. It is important to notice the changes in period for both pendulums. The main pendulum's period is shortened while the damping pendulum's period increases. This is noted in the phase relationship between the two plotted lines which is consistent with the theory developed by Frahm.



Graph 2.1 Tuned Pendulum Simulation

The second result of the simulation lies in the depiction of energy transfer between Pendulum 1 and its coupled damping pendulum. Examining the time study from 0 to 30 seconds shows Pendulum 1's oscillation reduction and the dramatic increase in the magnitude of the damping pendulum's sway. This is consistent with the results of Den Hartog's work and is indicative of the energy flow between the coupled systems. However, it is important that the damper not allow an uncontrolled energy transfer back to the initial body. The time period between 30 and 60 seconds shows such a transfer. Further examination reveals that the energy transfer proceeds in a cyclic fashion through the time study.

2.3 Significance To Crane Pendulation Damping

Although the simple pendulum system does produce a significant damping result, it is not considered a viable solution for crane payload damping. There are several reasons for this. The designer's ultimate goal in creating a damper is to reduce oscillation or vibration to improve an object's efficiency during use. A cable and mass hung below a crane's payload would interfere with efficient operation. Further, the pendulum damper, as noted in Graph 2.1, does not possess the ability to interrupt reversed energy transfer upon completing the initial damping. This would be an important design failure.

In light of the pendulum model's shortcomings, a reevaluation of the definition of an efficient damping design is needed. Focusing on the damper's ability to operate without failure is important. Evaluating the advantages of either reduced complexity or increased efficiency is also critical. Even justifying the use of semi-passive systems or active systems is significant. It is not realistic however, to ignore the damper's effect upon the operation of the crane. As noted above, the simple pendulum would have rendered the crane incapable of completing its tasks. A better design is needed to provide the same damping efficiency but from a smaller system operating without operation interference. Fortunately, tuned mass damping theory does not restrict the damping system to a specific format.

As noted earlier, tuned mass damping operates by coupling an oscillating object with a system "tuned" to oscillate at an optimum damping frequency. Taking this theory one step further, it is possible to replace the simple pendulum damper with any form of oscillating mass. A tuned mass damper design need only consider the effects of the mass ratio, natural frequency, and damping ratio required to achieve optimum results. The system could be engineered to include springs, rolling weights, a cantilevered rod, and other designs. The few restrictions include the coupling of the design and its ability to first absorb energy and then to control it for dissipation without interfering with the crane operation.

CHAPTER 3

CANTILEVERED ROD THEORY

Choosing a viable mechanism for tuned mass damping is considered an important aspect of designing a successful absorber. It has been noted that a number of alternatives are theoretically capable of achieving the desired damping. Thus an examination of the application is required.

The damping system is to be designed for application on mobile and tower construction cranes. Specifically the system is to be attached upon a spreader bar hung from the crane cable above the payload. As noted earlier, a proper design will accomplish damping with minor alteration to the crane's operation. To achieve damping under these constraints the ideal system would be light-weight, operate within a limited area, be easily adjusted, and require few moving parts.

Since the payload's oscillation period changes according to the effective cable length between the boom tip and payload, the damper's period must be adjustable. The period of spring and rolling mass systems are relatively hard to adjust. These designs also require special formats to achieve the long periods and large amplitudes required for damping (Fujinami, et al., 1990). In contrast, a cantilevered rod's period can be adjusted fairly easily and is theoretically capable of achieving the slow rate of oscillation. Further, a single cantilevered rod will oscillate throughout the X - Y dimensional plane. This means that a single rod could be a viable candidate in the damping of crane payload oscillations. Under these considerations the use of a cantilevered rod for testing purposes appears to be possible.

3.1 Cantilevered Rod Theory

By reviewing previous work in tuned mass absorbers detailed in chapter 2, it was learned that characteristics exist for achieving optimum damping. These include the ratio of the absorber's internal damping, ζ_s, the ratio of absorber mass to payload mass, μ, and the absorber frequency optimization, equation 1. To design an absorber according to these constraints, it is important that the absorber's operation is understood.

To achieve slow rates of oscillation with cantilevered rods, the rod is typically designed to operate in compression with a lumped mass at the top, as depicted in Figure 3.1. This reduces the effects of gravity upon the period of oscillation. In contrast, a rod operating in tension with the damping mass at the bottom would have a lower frequency limit of

$\frac{1}{2\pi}\sqrt{\frac{G}{L}}$. This limit corresponds to the lowest frequency achievable with a simple cantilevered rod acting in tension. To match the crane's frequency using this design, the rod length, L, would have to match or be greater than the length of the crane cable and thus interfere with the crane operation.

For the purpose of testing, the design will be similar to Figure 3.1.

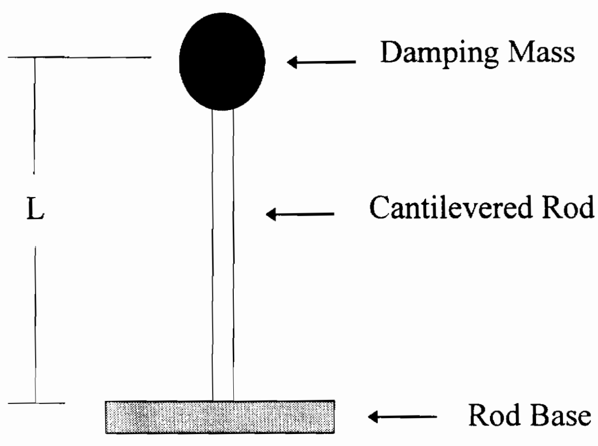


Figure 3.1 Cantilevered Rod In Compression

According to James et al. (James, et al., 1990) the rod depicted in Figure 3.1 will oscillate with a frequency of

Equation 4 $$f_n = \frac{1}{2}\sqrt{\frac{G}{W}\left(\frac{EI\pi^2}{32L^3} - \frac{W}{8L}\right)}$$ Cantilevered Rod Frequency

in which E is the rod's modulus of elasticity, I is the rod's moment of inertia, L is the rod length, and W is the weight of the damping mass.

From this equation, it can be seen that several factors effect the period of oscillation. The first is the amount of weight to be used. The second is the length of the rod. The third is the stiffness factor, EI. For a circular cross section the stiffness factor is defined by

Equation 6 $$EI = E\left(.25\pi R^4\right)$$ Stiffness Factor

which reveals that both rod radius, R, and the modulus of elasticity, E, affect the frequency of oscillation.

The ability of the rod to operate without failure should also be considered in the rod's design. For a cantilevered rod operating in compression the critical buckling load (P_{cr}) should be examined. This is the load or weight at which the rod will no longer oscillate when displaced from center as defined by

Equation 7 $$P_{cr} = \frac{EI\pi^2}{4L^2}$$ Critical Buckling Load

A second consideration concerns the rod's ability to function under cyclic loading. Cyclic loading greatly reduces the allowable stress that a material can withstand. An evaluation of the material's endurance limit also known as the fatigue strength is thus suggested. The endurance limit is a measure of the maximum allowable stress a material specimen can withstand without failure over the duration of 1000 stress cycles. As an example the recommended design stress for A36 steel under static loading is 22 ksi. The same specimen under cyclic loading is only recommended for 13 ksi loads (Budinski, 1992) With this in consideration, the design must meet a balance between rod character and damping weight necessary to achieve the optimum damping frequency without failure.

3.2 Cantilevered Rod Design

Keeping the critical buckling load and material endurance limit in mind, the cantilevered rod is designed to oscillate within a range of frequencies for an application as defined by tuned mass damping theory. In operation however, only a few constraints can be manipulated to achieve the optimum frequency. Thus the design process must evaluate the rod choice and damping weight to be used.

Material choice is an important aspect. It is rather obvious that some materials will be stronger in compression than others. Further examination reveals however, that materials like composites have higher strength to weight ratios. Polymers have high strength to weight ratios and higher rates of internal damping which would benefit the rod's ability to operate as a tuned mass damper. The material's ability to function under cyclic loading without failure is another important factor. Modulus of elasticity which effects the materials ability to oscillate can also vary greatly between materials (Budinski, 1992). Appendix "B" is a collection of charts detailing a few material properties.

The damping mass is another component that must be chosen prior to operation. Typically, a mass ratio is evaluated according to the damper's application. It is important that the mass ratio be capable of achieving damping without severely affecting the crane operation. A mass ratio of 1 to 2% of the average payload would be considered sufficient

(Bachmann, 1992). This value could be raised to 5 to 10% to reduce the amplitude of the absorber's oscillations during operation and the corresponding fatigue. A concrete placement operation provides a good example. The conveyance of 2 CY buckets of concrete averages 3628 Kg. Knowing this, the designer would model the damper with a mass between 36.3 and 363 Kg.

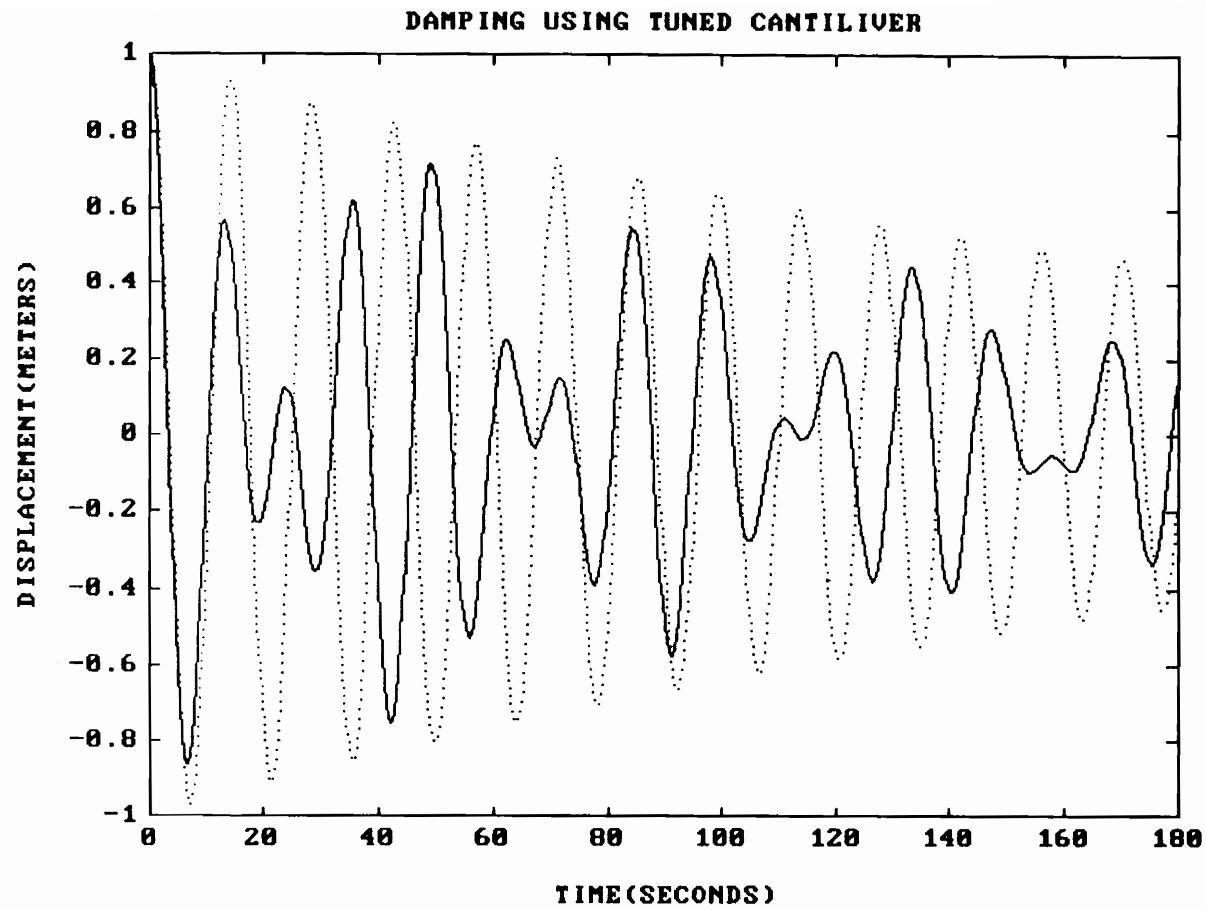
Having defined both rod material and damping mass, the designer must now choose the rod radius and appropriate rod lengths. Using a smaller rod radius allows the designer to achieve longer periods with shorter rods. This would reduce the damper's effect upon the crane operation. However, the rod radius and length must be chosen carefully to not exceed the critical buckling load.

3.3 Cantilevered Rod Damping

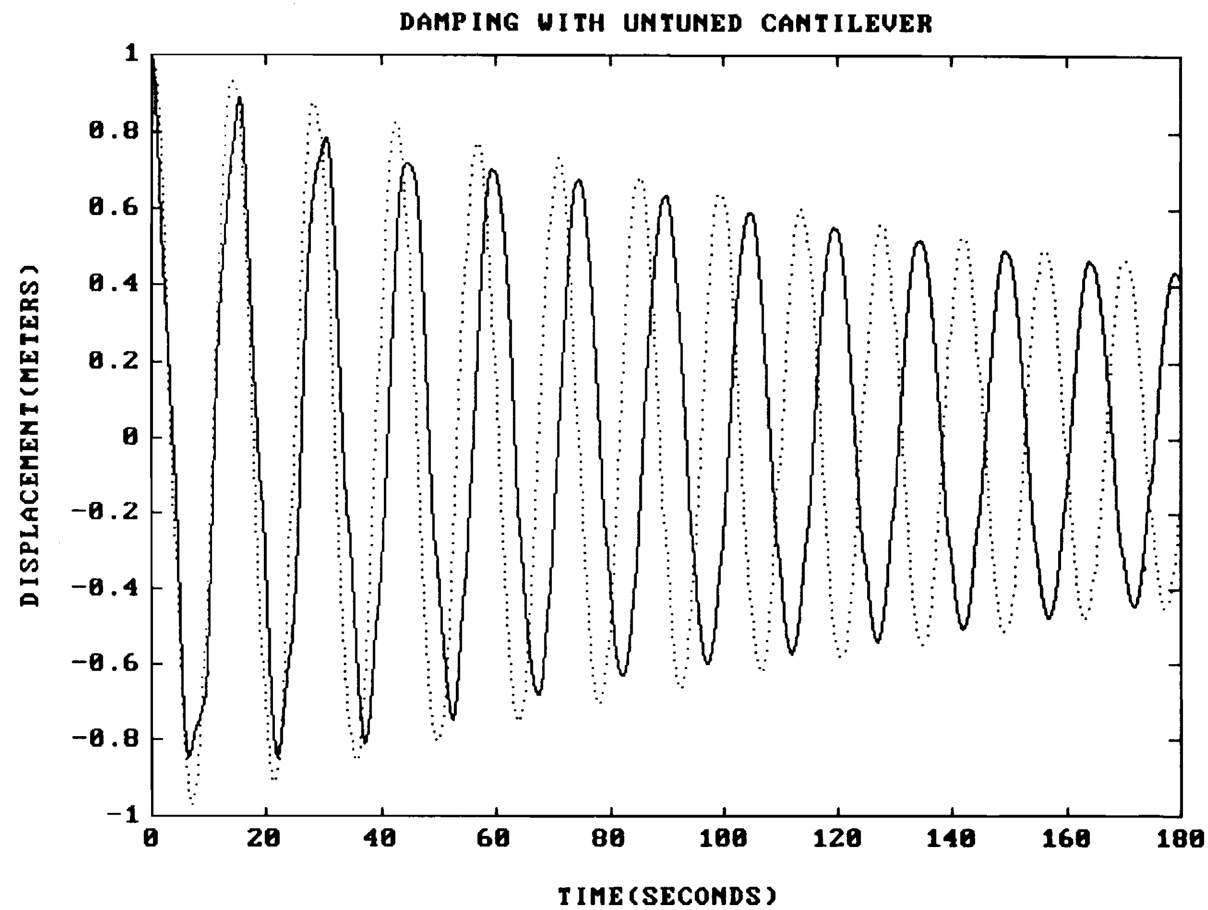
To evaluate the use of cantilevered rods as tuned mass dampers, a set of computer simulations were run on the program Matlab. These tests are similar to that of Graph 2.1. Here the frequency equations for a cantilevered rod as defined by Jean - Guy Beliveau (Beliveau, 1995) were substituted for the equations of the damping pendulum previously discussed. The results can be seen as Graphs 3.1 and 3.2. An example of the Matlab code has been included in Appendix A.

Graph 3.1 depicts the motion of a cantilevered rod tuned to oscillate at the pendulum frequency. The simulation suggests that the result will be very similar to that of a tuned pendulum. The dramatic increase in motion for the damping rod and the decrease in motion of the pendulum at 30 seconds is noted. Once again a decrease in the pendulum period and increase in the damper's period are represented.

Graph 3.2 depicts the interaction of the pendulum and a shorter damping rod tuned to a shorter period. In this simulation little to no damping exists and the rod eventually oscillates out of phase with the pendulum. This reveals the importance of proper tuning to achieve the desired damping effect.



Graph 3.1 Tuned Cantilever Simulation



Graph 3.2 Non-tuned Cantilever Simulation

CHAPTER 4

CANTILEVERED ROD EXPERIMENT

It is the intent of this chapter to develop a cantilevered damping rod model for testing. The model will take into account the design considerations required for application on a mobile construction crane. Through this testing, the physical characteristics of the cantilevered rod can be evaluated and compared to values predicted using the equations developed in chapter 3.

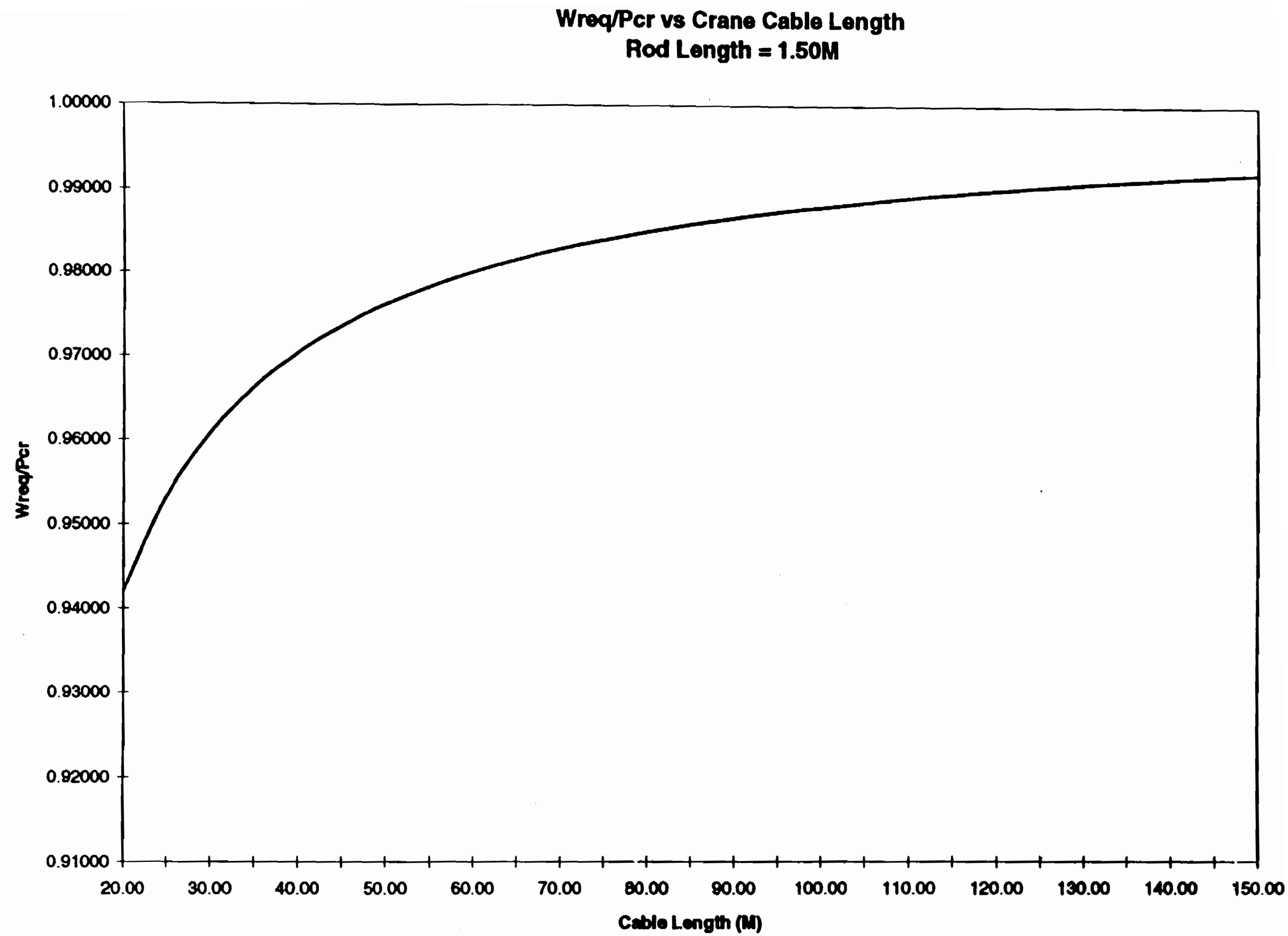
4.1 Model Design Considerations

As discussed in chapter 3, the design of the cantilevered rod damper must begin with an evaluation of the construction crane application. Most construction cranes incorporate extendible booms ranging from 20 to 150 meters. Longer lengths can be found on tower cranes. From the boom is extended a single woven steel cable from which a spreader bar is hung. This bar allows for multiple attachment points for stabilization of the payload during conveyance.

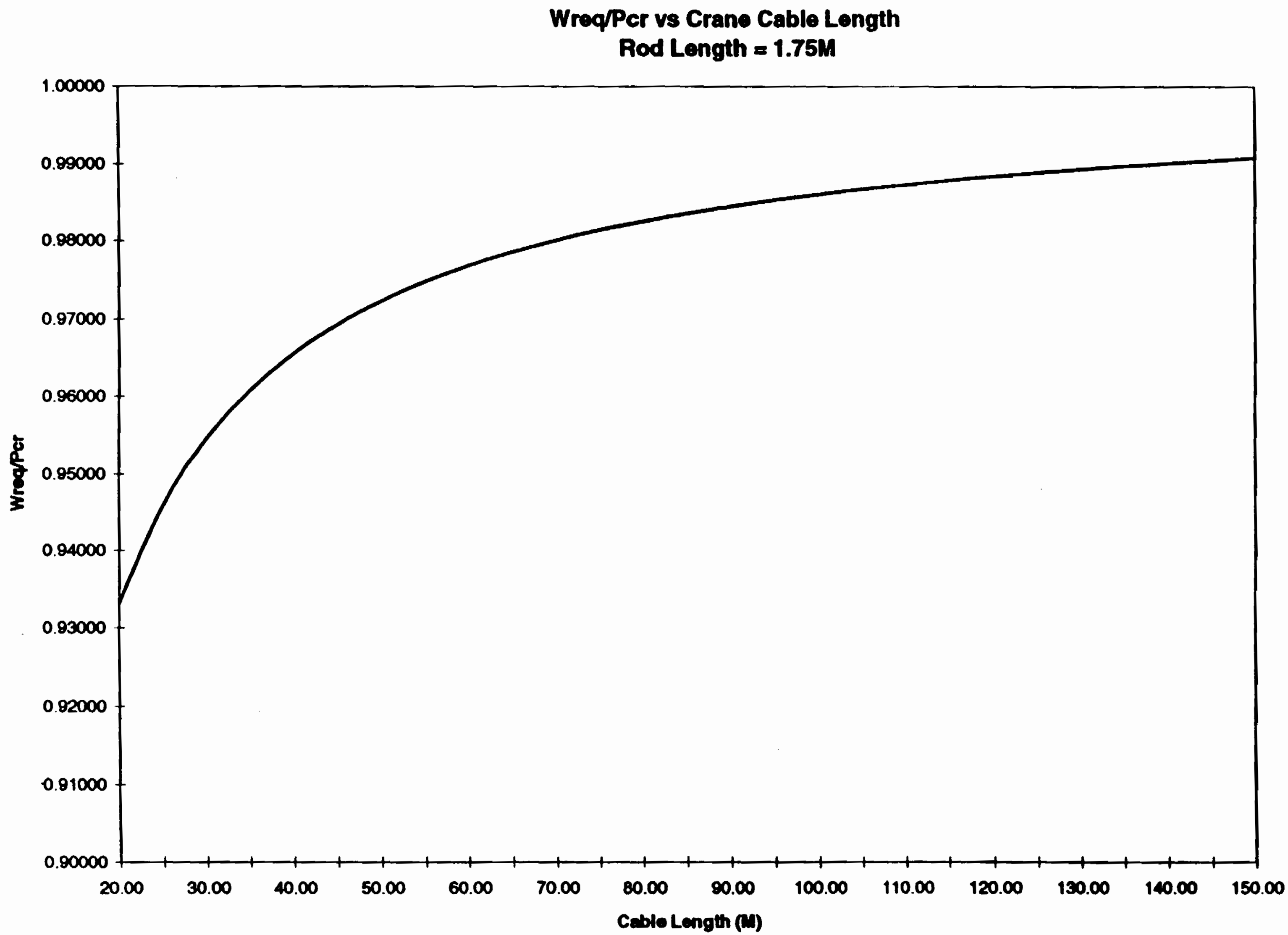
Although the source of payload pendulations can be attributed to a number of factors, the period of the oscillation is easily generated. Typically, such factors as boom dynamics and rotation of the crane base from soft ground are ignored to simplify the analysis (Ruddy, 1994). Under these constraints the crane cable and payload act much like a simple pendulum. The frequency of oscillation thus becomes $\frac{1}{2\pi}\sqrt{\frac{G}{L}}$, where G is gravitational acceleration and L is effective cable length. Using this equation crane frequencies for 20 to 150 M cable lengths becomes .11 to .04 Hz.

T ≅ 9sec .. 25 sec

An evaluation of the rod and weight characteristics required to reach crane payload oscillation frequencies indicates that the critical buckling load could prove to be a problem if the damping rod is to operate in compression as designed. For example, Graphs 4.1 and 4.2 show the comparison of the weight required to achieve the defined frequencies as compared to the critical buckling load, W_{req}/P_{cr}, on the vertical axis. Weights are varied on rods of constant dimension to achieve the frequency detailed on the horizontal axis. From this it can be seen that the ratio of weight required to critical buckling load is within 90%. However, the two graphs suggest that longer rod lengths reduce this ratio.



Graph 4.1 Critical Buckling Load Analysis



Graph 4.2 Critical Buckling Load Analysis

industry. Because it is used for many applications, rods of various dimensions and configurations are common. Although materials like composites and polymers have greater strength to weight ratios, steel is stronger in compression and maintains a higher endurance limit under cyclic loading. Polymers and composites, in contrast, are highly susceptible to fatigue in cyclic loading (Budinski, 1992). Aluminum a possible alternative to steel is also considered inappropriate. Although it is lighter than steel, it is more susceptible to fatigue damage in cyclic loading.

4.2 Experiment Design

To test the validity of the equations defined by James et al. a series of experiments was conducted. Figure 4.1 depicts the experimental design.

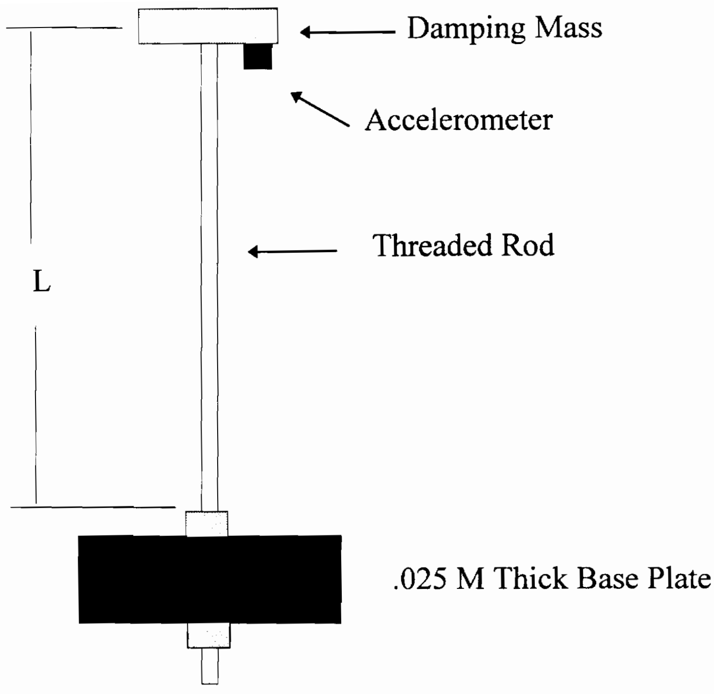


Figure 4.1 Cantilevered Rod Experiment

The design incorporates a threaded steel rod. The manufacturer quoted the modulus of elasticity to be 210 Newtons / Meter Squared (N/M^2). The major radius was found to be .00476 Meters (M) with pitch radius of .00425 M. Frequency predictions are based upon the .00373 M root radius. Root radius is defined as the radius of the rod if the threading had been removed.

The rod was cantilevered by creating a static connection on the lower end. This was achieved by passing the rod through a threaded hole drilled in a .025 M thick steel base plate. The connection was secured with nuts and washers.

Three damping masses ranging from 4.075 to 12.06 Kg were used to conduct a series of tests. From each mass was glued a Kistler Model 8602A500 accelerometer. This device was operated through a Kistler Model 5004 dual mode amplifier. Data was taken at a rate of .01 samples per second using a Dash-8 data acquisition board and the program Labtech Notebook produced by the Laboratory Technologies Corporation.

The experiment was conducted in a simple repeated procedure.

1. The rod, base plate, and damping masses were secured using the nuts and washers.
2. The effective rod length was measured from the center of the mass to the top of the base plate.
3. The masses were manually set into oscillation.
4. The accelerometer's output was sampled by the computer for a period of 8 seconds.
5. The procedure was repeated for an effective rod length which was decreased .005 M.

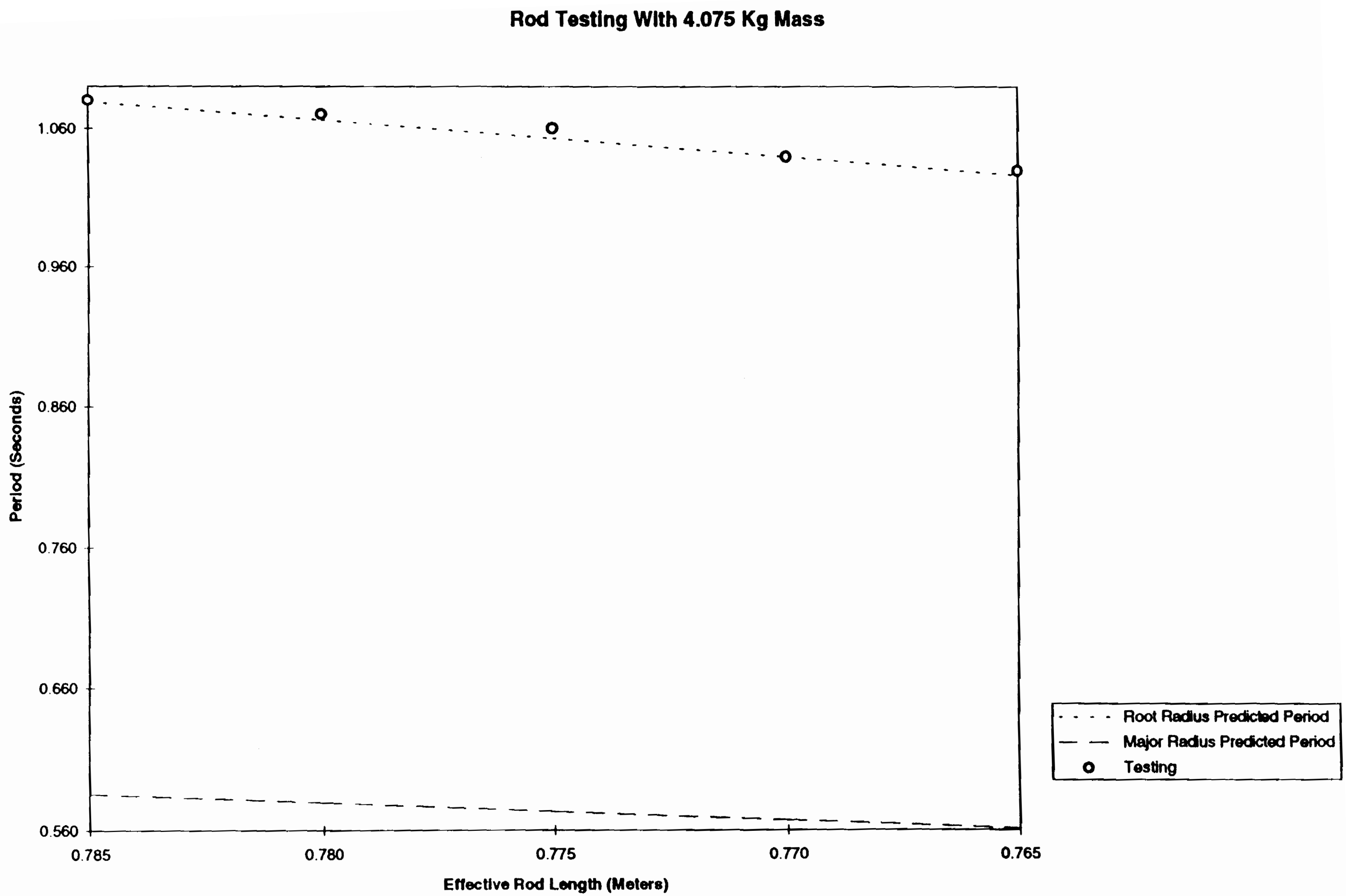
4.3 Experimental Analysis

Data sampled during the experiments was analyzed to evaluate several parameters. The first was defined by the natural period of the rod and damping mass combination in physical testing. The data was imported into the spreadsheet program Microsoft Excel for the analysis. The periods as derived from the oscillation samples were then plotted with predicted results computed from the equations derived by Beliveau (Beliveau, 1995) and James et al. (James, et al., 1990). The results appear in graphs 4.3, 4.4, and 4.5 which are further described in the next section of this thesis.

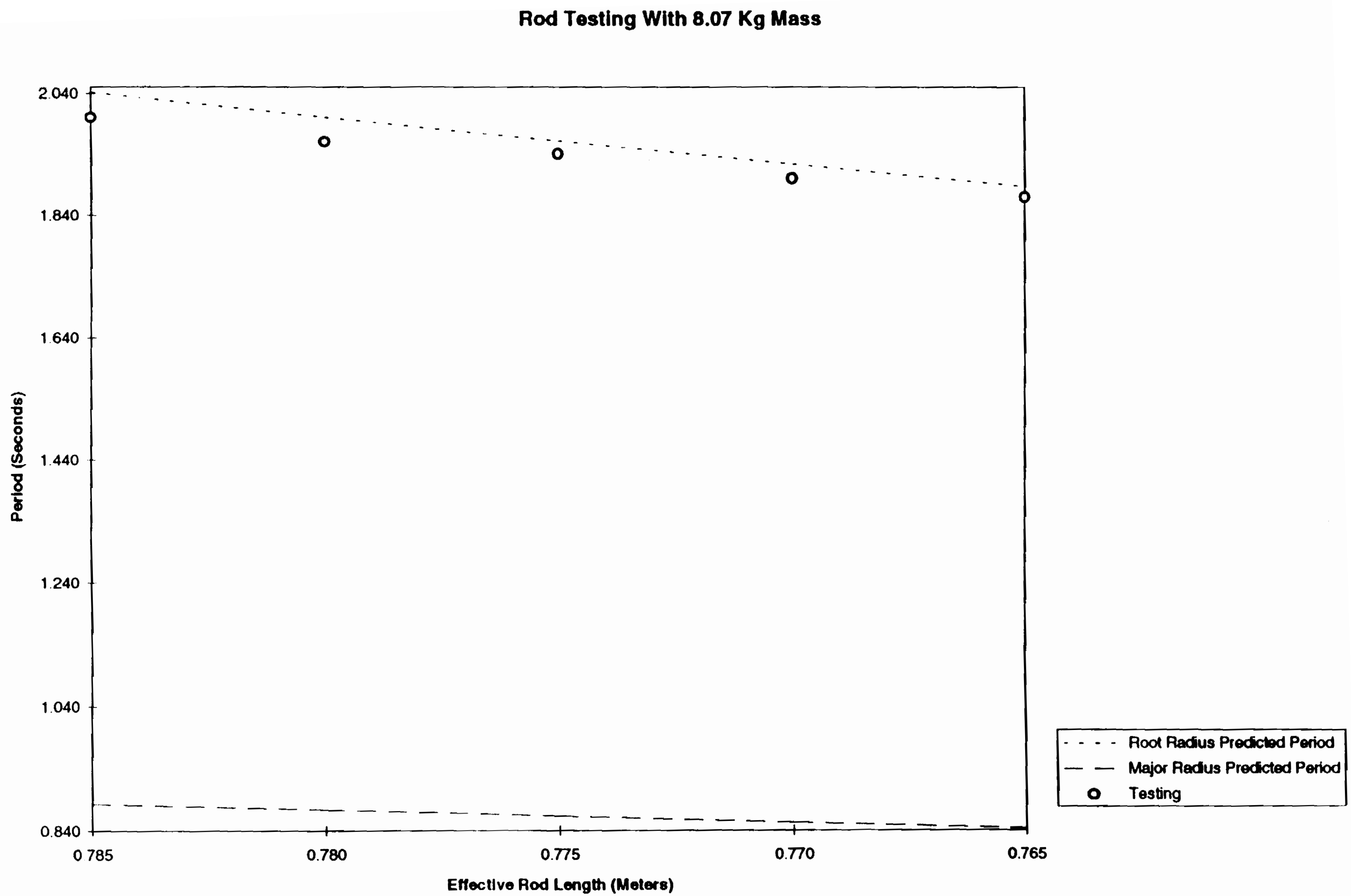
The second evaluation was used to analyze the ratio of internal damping, ζ, for the rod and damping mass combinations. This was accomplished using Logarithmic Decrement Technique as defined by the equation below.

Equation 8 $$\zeta = \frac{\ln \frac{(x(t))}{(x(t+nt))}}{2\pi N}$$ Ratio Of Internal Damping

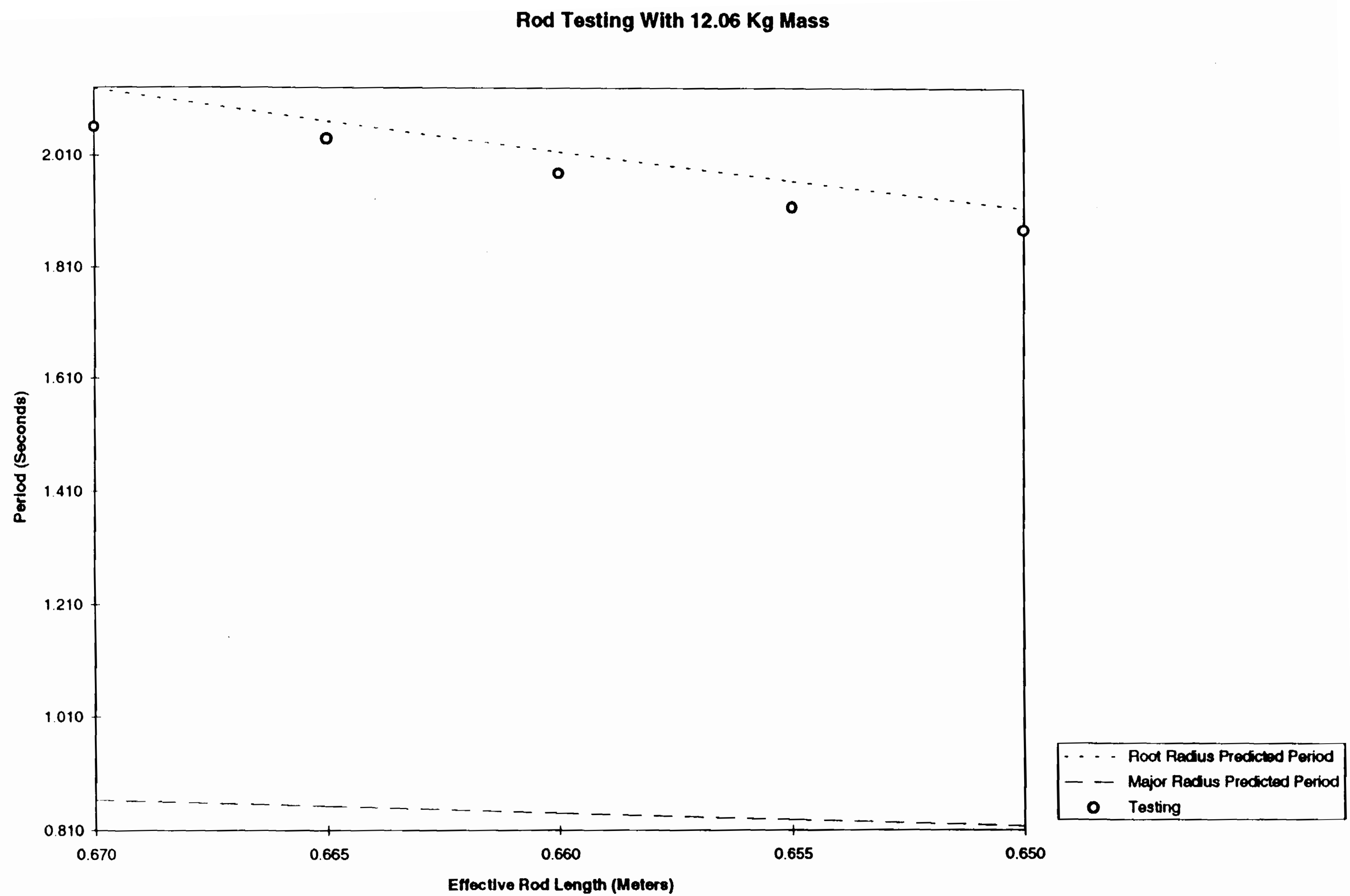
In the equation x(t) corresponds to the first oscillation peak over a defined number of cycle peaks (N). x(t + nt) corresponds to the oscillation peak at the end of N cycle peaks. The value of data sampled by piezometric accelerometers decreases over time. To compensate for this, the mean value of the first oscillation cycle and the last oscillation cycle were compared. With this comparison the last cycle could be adjusted to produce the correct peak value, x(t + nt).



Graph 4.3 Experiment 1 Results



Graph 4.4 Experiment 1 Results



Graph 4.5 Experiment 1 Results

4.4 Experimental Results

Graphs 4.3, 4.4, and 4.5 provide an indication of the equations' validity. All of the measured periods closely follow the values as predicted from the root radius. Their proximity to the root radius values, as opposed to the values for the major radius, reflect the minor effect of the threads upon the rod's oscillations. It can thus be assumed that the threading will produce a minor, if not negligible, effect throughout future damping tests.

The testing also reveals the damping mass's effects upon rod oscillations. Rod tests with the 4.076 Kg mass follow the root radius predictions very closely. The data corresponding to the heavier masses are plotted at values with increasingly shorter periods. These shorter periods would correspond to either a shorter rod length than measured or a rod of greater stiffness than predicted

Error in rod length evaluation could be traced to human error in the measuring process. The shorter rod length can also be attributed to the reaction of the nut and washers used to secure the rod to the base plate. The predicted results are computed from the rod length as measured from the center of the mass to the top of the base plate. It is possible that the nut and washers could be shortening the effective length of the rod by increasing the base plate's effective thickness.

Greater rod stiffness can be attributed to several sources. The predicted stiffness factor, EI, is computed from the rod's root radius. The threads could be adding to the stiffness by increasing the rod's effective radius. A concept known as "seating" suggested by Jean - Guy Beliveau could also contribute to the increased stiffness. In this theory the static qualities of the threaded connection could be increased by the compressive force of the increasing damping masses. It is noted that the difference between predicted values and testing values increases with increasing mass.

The limited scattering of the testing results is caused by both human error in the measurement process and noise caused by the sampling equipment. This is recognized as small measurement deviations along the overall sampling path.

CHAPTER 5

TUNED CANTILEVERED ROD

The previous chapters have been dedicated to the theoretical design of a cantilevered rod to be operated as a tuned mass damper. This chapter will detail the design of the cantilevered rod damper for physical testing. Tuned mass damping theory will be used to define design parameters for application on a scaled crane model. Through this testing the chapter intends to confirm the concept of tuned mass damping theory for application on construction cranes.

5.1 Pendulum Model A

The design of a tuned mass damper begins with an evaluation of the structure to be damped. The scaled crane model consists of a pendulum system incorporating 4 woven 1/4 Inch steel cables, a spreader plate, and payload mass of 108 Kg. Pendulum model A is presented in Figure 5.1 with the damper as it would be attached for operation.

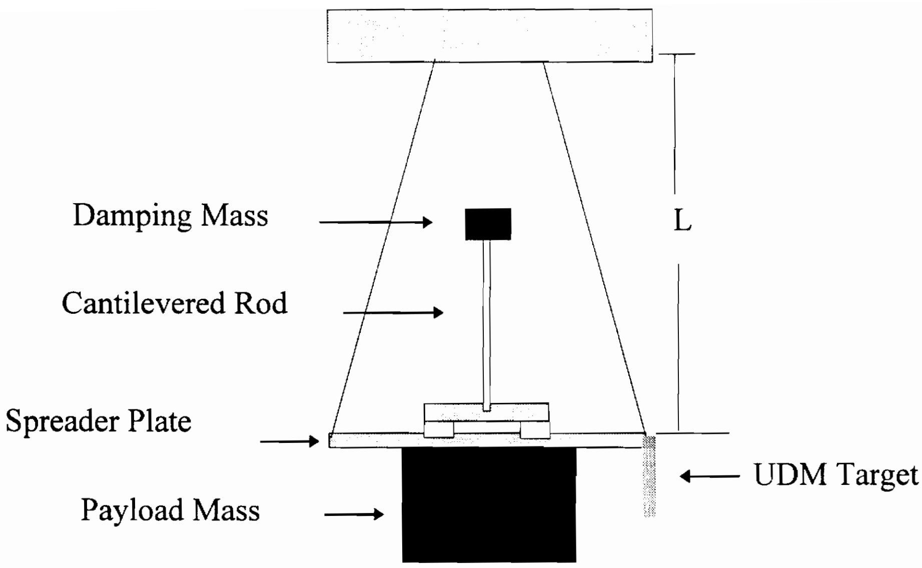


Figure 5.1 Pendulum Model A

The crane pendulum cables were secured from a static steel girder to create an effective length, L, of 1.26 Meters. The cantilevered rod, damping mass, and base plate were coupled directly to the spreader plate using nuts and bolts. The connection was secured to prevent motion or deflection between the pendulum system and the damping system. Data acquisition sensors included a 1 dimensional accelerometer epoxied to the bottom of the damping mass and an ultra sonic distance meter (UDM) calibrated to read displacements from the spreader plate position.

Tests were conducted to confirm the properties of pendulum model A with the damping system removed. The total payload was defined as 122 Kg. This included the mass of the spreader plate, base plate, and payload mass. UDM data was sampled to acquire the pendulum's period and rate of internal damping. The pendulum was released from an initial displacement from center of .16 M or a displacement angle of 7.24 Degrees. This large displacement was used to exaggerate the effects of the pendulum motion. Maximum allowable crane payload sway in operation has been defined as 3.5 Degrees (Ruddy, 1994). The testing results indicated that pendulum model 1 had a period of 2.27 seconds and approximate frequency of .44 Hz. Through evaluating the decay rate of oscillations with logarithmic decrement technique, the internal damping was calculated to be 1.32% using Equation 8.

5.2 Tuned Mass Damping Rod Design

With the character of the pendulum understood, the tuned mass damper was designed. As suggested in section 2.1, the damper mass is typically the first parameter chosen in the design process. The damping mass in this testing was defined to be 2.0 Kg. This resulted in a mass ratio, μ, of 1.6% which is within the 1 to 2% ratio common to other research. Although larger mass ratio's typically have smaller damping displacements, smaller mass ratios use higher optimized frequencies. Lower optimized frequencies are difficult to achieve. As mentioned earlier, few spring systems are capable of achieving long periods of oscillation. In the case of the cantilevered rod, the rod length must increase to reach longer periods. This increased rod length produces greater rod interference during crane operation while placing the rod closer to the critical buckling load.

Knowing the character of the structure's oscillations and the damping ratio to be used, the optimum damping frequency and rate of internal damping, ζ, were calculated with the design equations. These two parameters were evaluated using the equations defined by the tuned mass damping theory detailed in chapter 2. An optimum damping frequency of .43 Hz and 7.6% internal damping were calculated.

In chapter 4 steel was chosen as the cantilevered rod material. Using the modulus of elasticity defined by the manufacturer, several rod lengths and radii were evaluated using the cantilevered rod frequency equations. Ultimately a .00635 M diameter threaded rod with root radius of .00254 M was chosen for testing. This rod was commonly available and inexpensive. The small radius also decreased the length required to achieve the optimized frequencies.

The rod was then tested to confirm the frequency of oscillation. Tests were completed using the procedure defined in chapter 4. Analysis of accelerometer data indicated an effective rod length of .668 M was required to achieve the optimized damping frequency. Frequencies were also evaluated at shorter and longer rod lengths for further testing at various damper and pendulum configurations.

An evaluation of internal damping was also conducted from the accelerometer output sampled during the rod analysis. From this testing the rod was found to have .4% internal damping on average. This value has important repercussions in the testing and results. .4% is lower than both the optimized internal damping defined through calculation and pendulum model A's internal damping calculated through testing. The researcher felt that the rate of internal damping would prevent the damper from achieving optimum damping results. However, testing was conducted with the model to observe the energy flow between the pendulum and damper. It was predicted that a cyclic flow of energy and resultant change in oscillation amplitude would be observed. The damper would not however, be capable of controlled energy dissipation. This would be comparable to the tuned pendulum model presented in chapter 2 and Graph 2.1. Parameters for pendulum model A and it's damper have been compiled in Table 5.1.

Pendulum Model A			**Cantilevered Damping Rod A**		
Payload Mass	=	122 Kg	Damping Mass	=	2.0 Kg
Frequency	=	.44 Hz	Optimum Frequency	=	.43 Hz
Internal Damping	=	1.32 %	Optimum Internal Damping	=	7.6 %
Release Displacement	=	7.24 Degrees	Actual Internal Damping	=	0.04 %

Table 5.1 Pendulum Model A and Damper

5.3 Pendulum Model A Experiment

To test the cantilevered rod's ability to damp the oscillations of pendulum model A, a testing procedure was followed. The procedure was conducted in the repeated format defined below.

1. The damping mass was secured to the isolated cantilevered rod and tested to confirm the period and frequency predicted in earlier testing.
2. The cantilevered rod system was attached to pendulum model A's spreader plate using nuts and bolts.
3. The pendulum was manually held at .16 M of initial displacement.
4. The pendulum was released.
5. UDM and accelerometer data was sampled at a rate of .02 seconds per sample for 60 seconds.
6. The cantilevered rod damper was removed to begin the testing procedure at a new rod frequency.

A test was conducted at the optimized damping frequency. Tests above and below the optimized frequency were then completed. These frequencies were defined by increasing and decreasing the effective rod length.

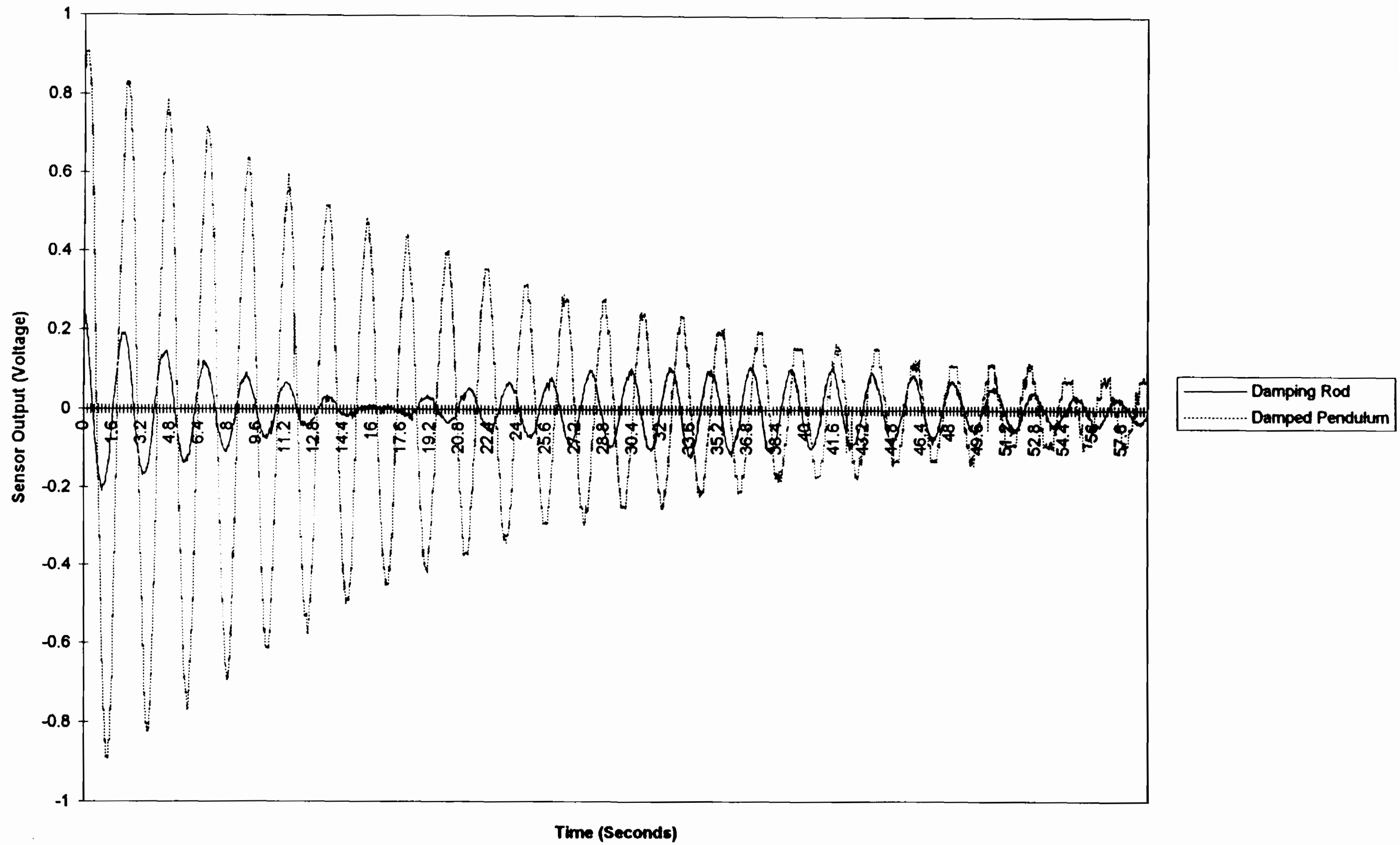
5.4 Pendulum Model A Testing Results

Upon completing the testing, data was evaluated using Microsoft Excel to plot sensor output. Data is left in terms of voltage to evaluate the oscillations of the pendulum and damper during operation. Several reactions were predicted to appear in these graphs. If the damper worked, a cyclic energy transfer should have been noted between the pendulum and cantilevered rod. This would appear as a cyclic increase and decrease in both damper mass acceleration and pendulum oscillation amplitude. The damper mass should have also shown greater accelerations with the large damper oscillation amplitudes associated with tuned mass damping.

Graphs 5.1 and 5.2 depict data collected in testing with an optimized cantilevered rod damper frequency. Graph 5.1 shows the voltage output of the sensors. The accelerometer's output from the cantilevered rod does not suggest the high rates of acceleration that would correspond to large amplitude oscillations. It is also clear that while the damper's oscillations increase and decrease cyclically, the pendulum does not undergo such a reaction. This is confirmed in Graph 5.2. Here a comparison is made between the oscillations of a similar undamped pendulum and the pendulum as damped by the cantilevered rod. This graph does not indicate the cyclic decrease in oscillation amplitude predicted in the theoretical simulation or the tuned damping analysis. It appears that the damper had little to no effect upon pendulum model A.

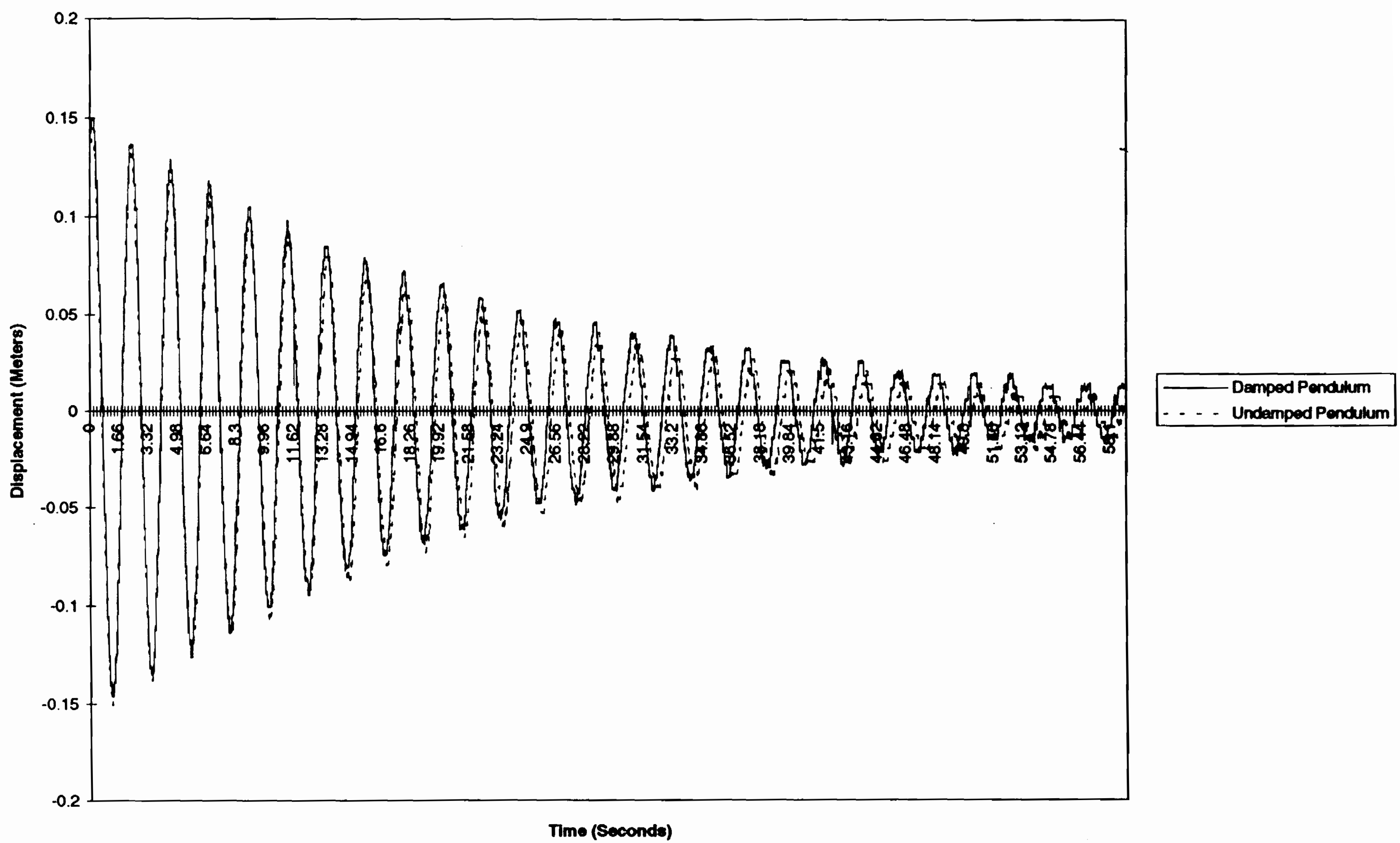
Tests conducted at frequencies above and below the optimized frequency produced similar results. The attached tuned mass damper does not effect pendulum model A as predicted by tuned mass damping theory. Graphs 5.3 and 5.4 depict the sensor output and pendulum oscillation amplitudes achieved in testing with a rod frequency greater than the optimized frequency. Again no noticeable damping was experienced.

1.67% Mass Ratio Optimized Rod Frequency



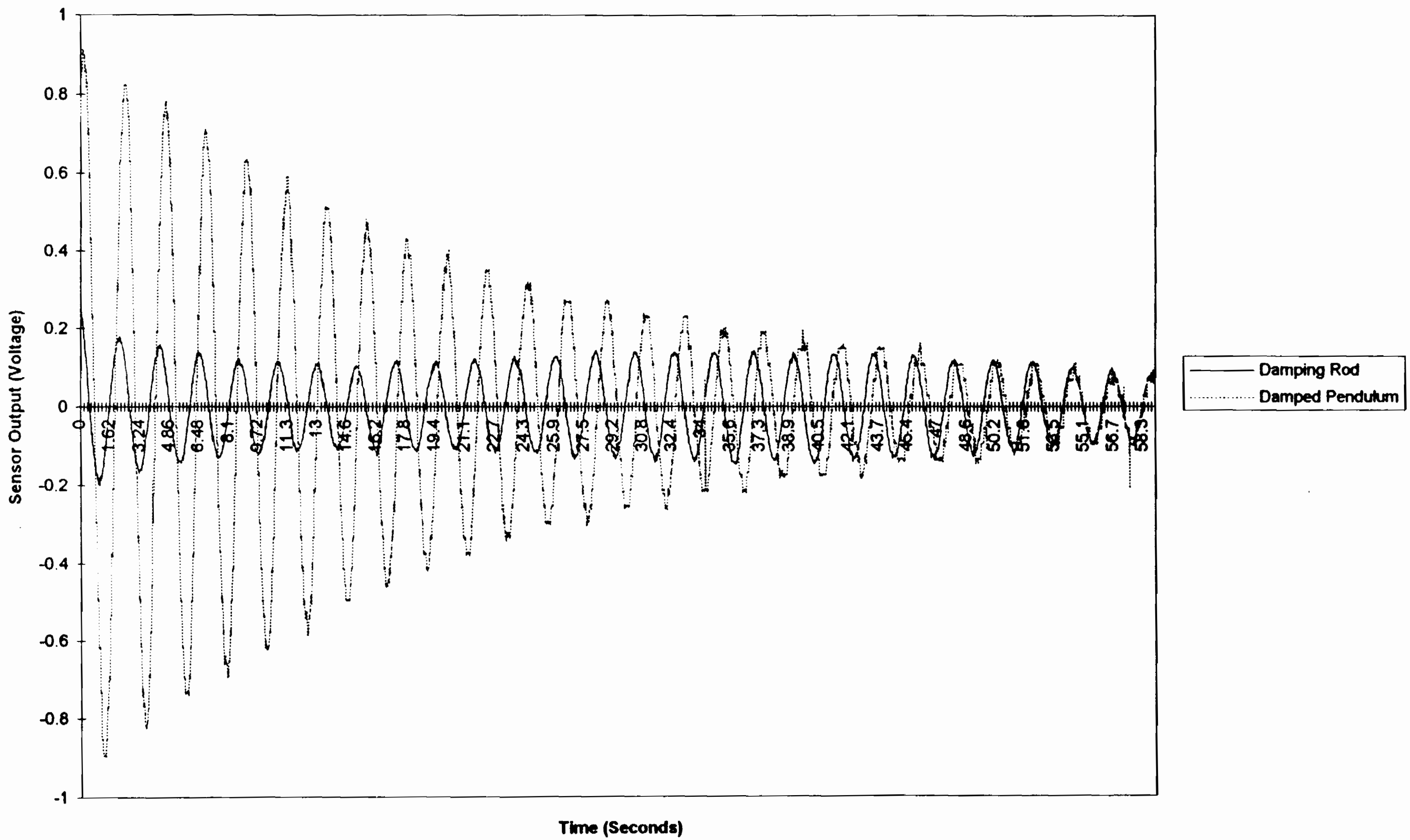
Graph 5.1 Pendulum Model A Testing

1.67% Mass Ratio Optimized Rod Frequency



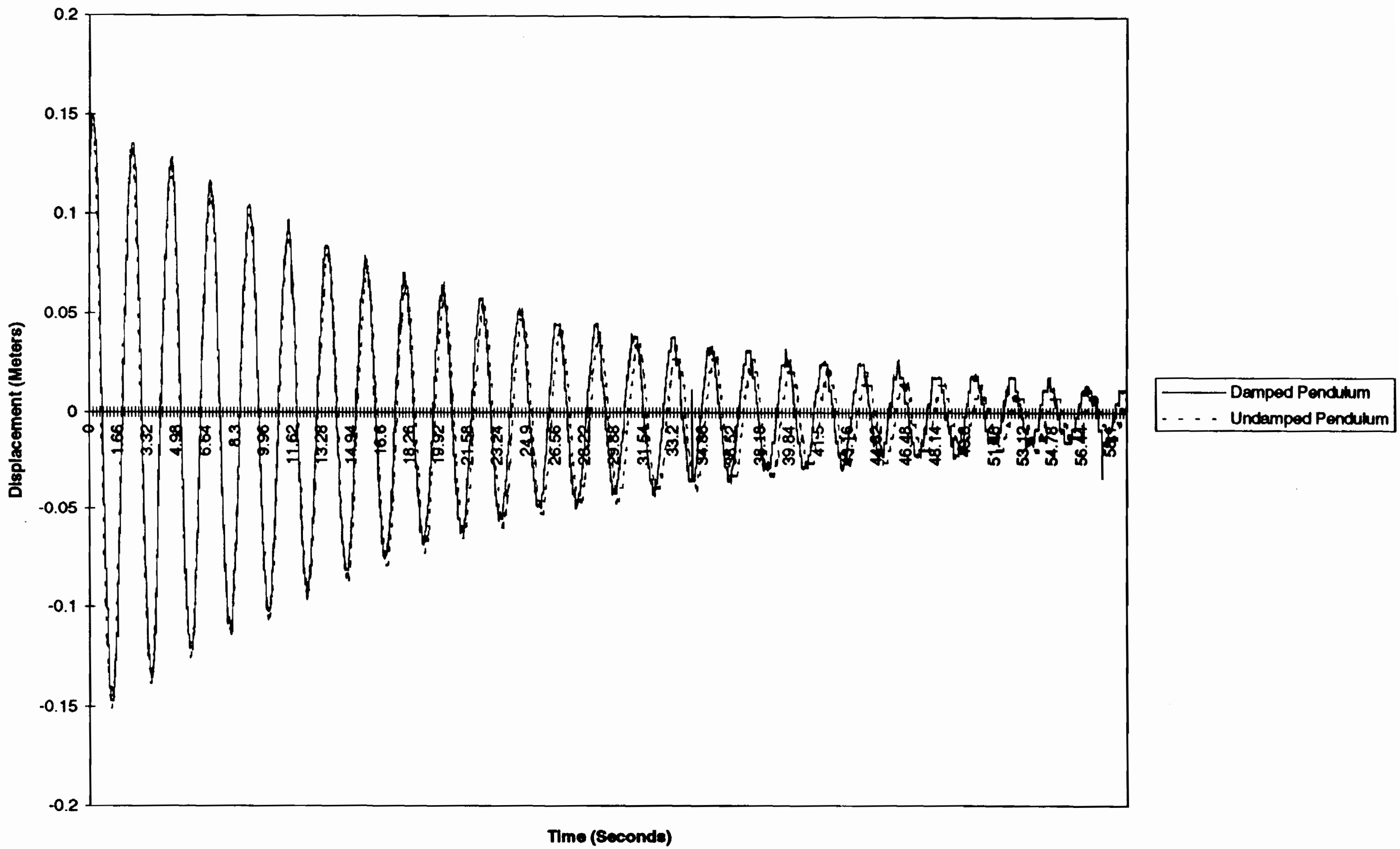
Graph 5.2 Pendulum Model A Testing

1.67% Mass Ratio Higher Rod Frequency



Graph 5.3 Pendulum Model A Testing

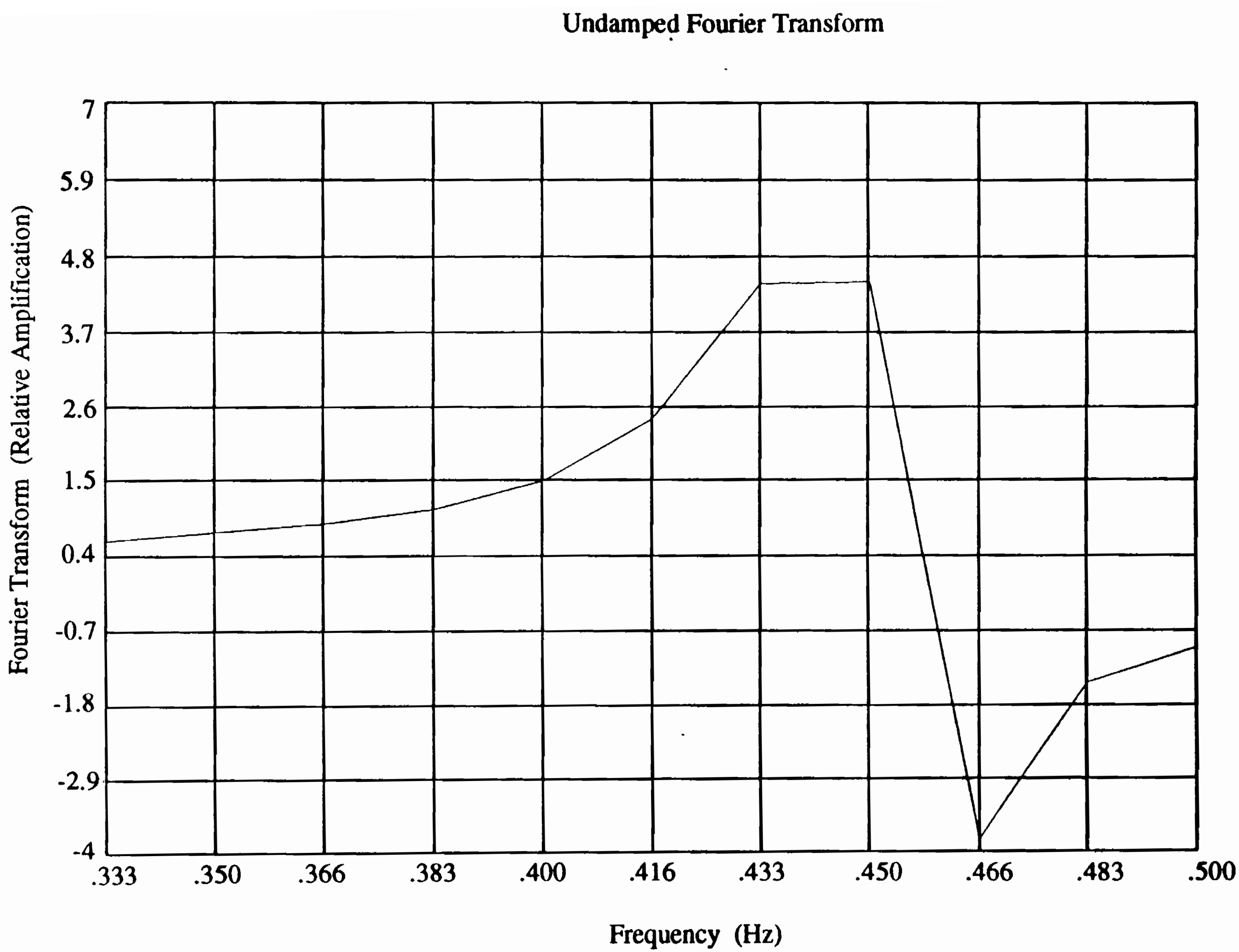
1.67% Mass Ratio Higher Rod Frequency



Graph 5.4 Pendulum Model A Testing

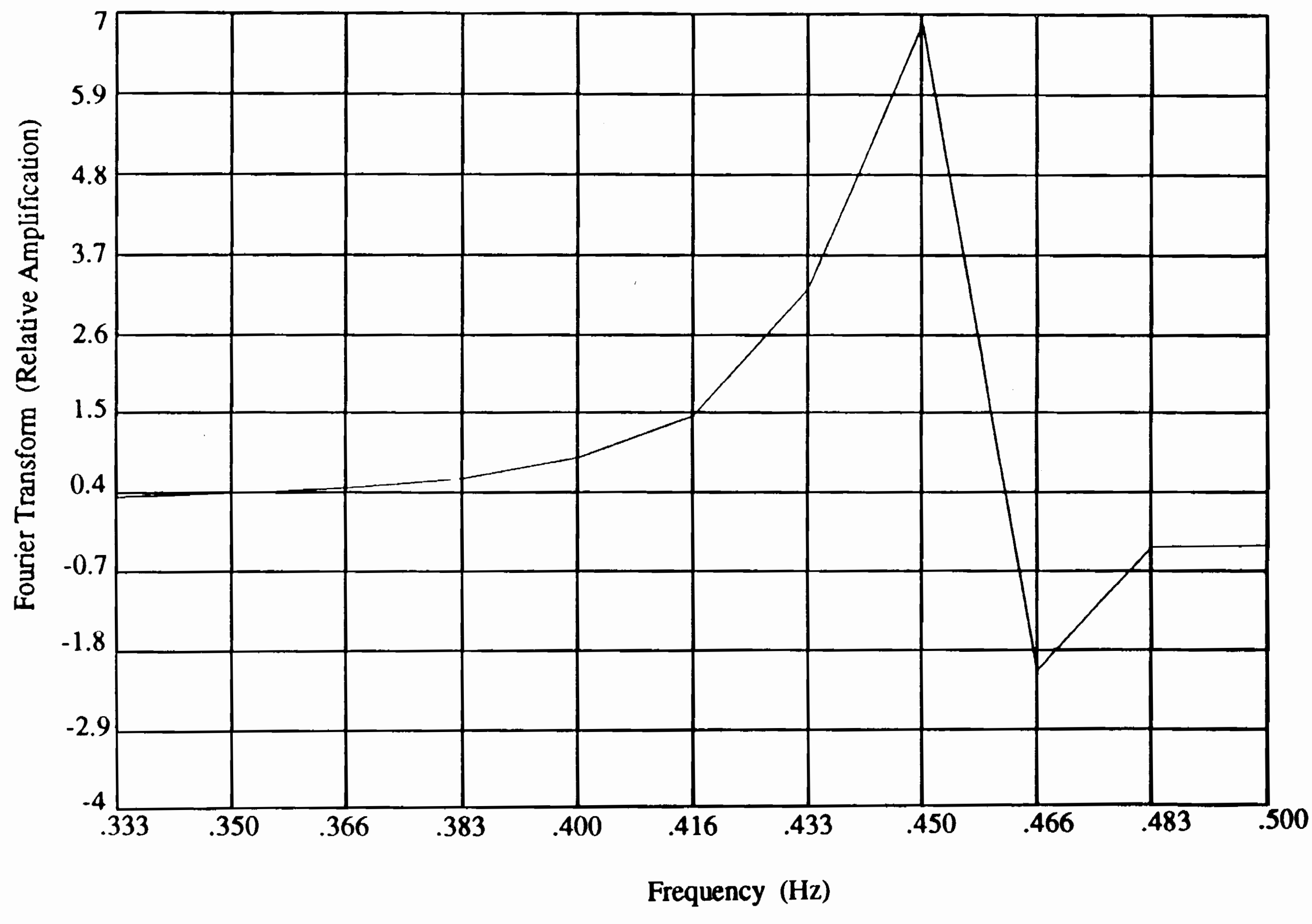
As a second test, Fourier transforms of the pendulum oscillations were calculated. Fourier transforms produce a relative amplification value from the frequency energy generated by a system in free vibration following a single excitation force. This is similar to the power amplification curve produced by Bachmann et al. for a system reacting to excitation forces applied sinusoidaly (Bachmann, et al., 1995) . As described in chapter 2, a tuned mass damper will cause the single amplification peak of an undamped structure to split into two new frequencies with decreased amplification. Graph 5.5 represents the Fourier transform of the undamped pendulum model A. Graphs 5.6 and 5.7 represent the plots of the damped pendulum with an optimized damper and lower frequency (non-optimized) damper respectively. The testing equipment limited the rate of data sampling. Higher rates of measurement over a shorter time period would have provided a better resolution thus forming a smoother graph with more accurate interpretation of the frequency peaks.

An evaluation of Graphs 5.5, 5.6, and 5.7 gives an indication that tuned mass damping was not achieved. Although some sampling error is responsible for the data miscalculation in Graph 5.5, the plot is representative of the single peak associated with an undamped structure. Graphs 5.6 and 5.7 do not however produce the frequency split and amplitude reduction associated with tuned mass damping. While some discrepancy in peak amplification value is noted, this corresponds to differences in testing release displacements caused by operator error.



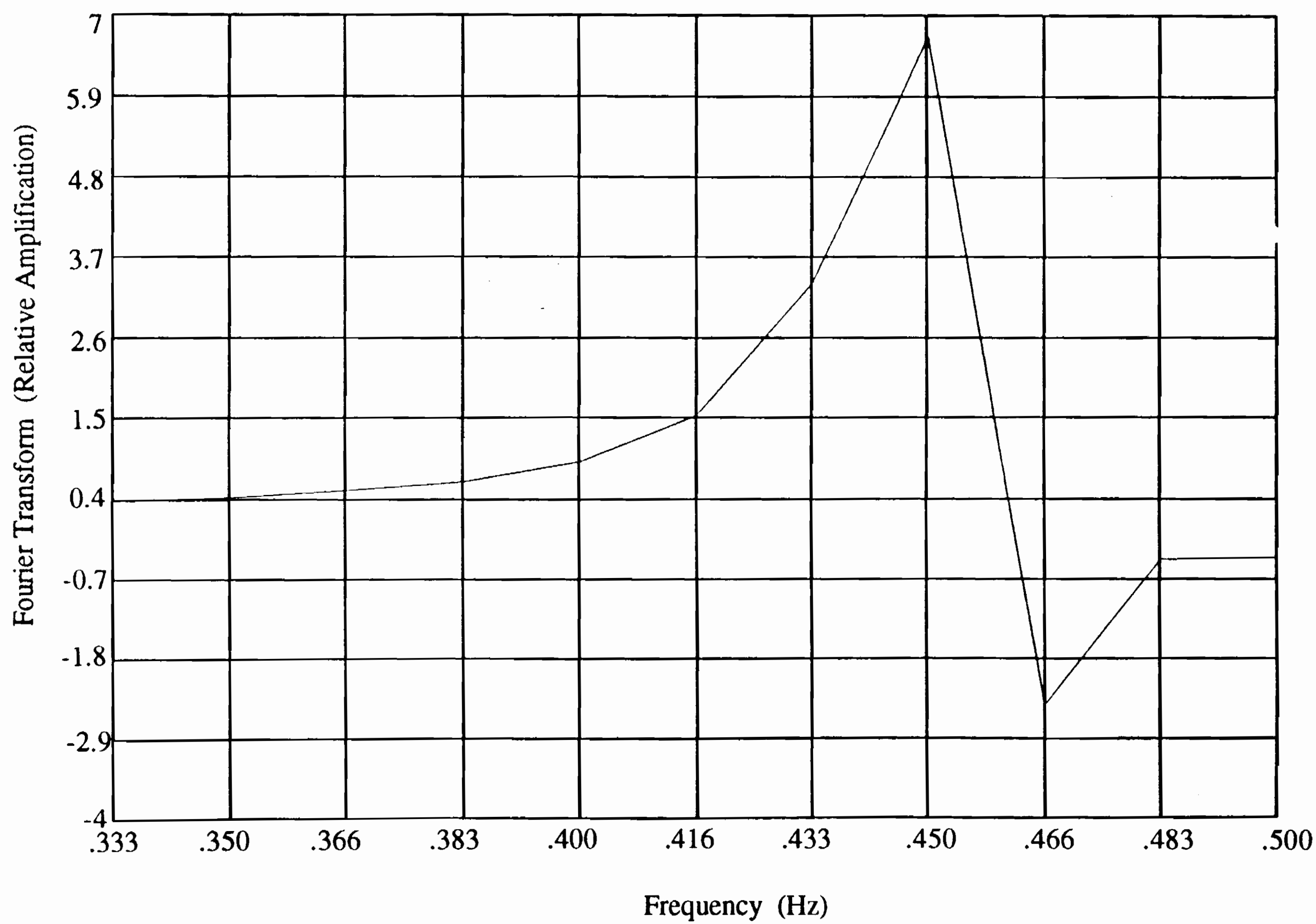
Graph 5.5 Pendulum Model A Testing

Optimized Rod Frequency Fourier Transform



Graph 5.6 Pendulum Model A Testing

Lower Rod Frequency Fourier Transform



Graph 5.7 Pendulum Model A Testing

To further test pendulum model A, alternative mass ratios and rod frequency values were tested. Using frequency values far above and below the optimized frequency did not produce a tuned mass damping effect. Likewise, tests conducted with mass ratios of 6.57% and 13% did not achieve the predicted effect. Throughout the testing, the same design and testing procedures were conducted. Care was taken to remove the effects of operator error in the testing procedure at each step. The results did not show any response to the changed frequencies and mass ratios.

5.5 Pendulum Model A Analysis

In evaluating the test results presented above, it appears that the cantilevered rod's rate of internal damping may have a greater effect than predicted. An examination of Graph 5.1 indicates that the damper's accelerations and corresponding oscillations were brought to rest at several points. It is possible that the pendulum is absorbing and dissipating the damper's oscillation energy. This would suggest that increasing the damper's rate of internal damping above the pendulum's could reverse the effect.

5.6 Pendulum Model B

To test the validity of the conclusions made from pendulum model A's analysis, a second model was constructed for testing. The model was kept in the same format except for the attachment design. This model incorporated cable diameters of 1/8 Inch and centralized attachment points. The model was tested as it appears in Figure 5.2.

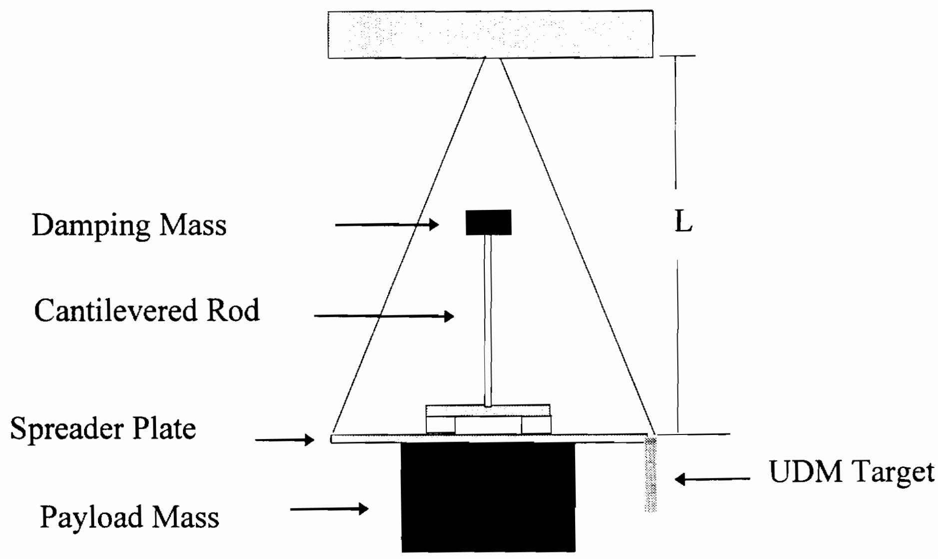


Figure 5.2 Pendulum Model B

Tests were conducted to confirm the properties of pendulum model B. With the damping system removed, data was sampled for oscillations from an initial displacement of .16 M or 7.24 Degrees. Analysis of UDM data indicated a period of 2.28 seconds and approximate frequency of .439 Hz. The increased period resulted from an increase in effective cable length, L, to 1.3 M. Analysis using logarithmic decrement technique indicated .02% internal damping.

The damping mass was once again chosen to be 2.0 Kg producing a mass ratio of 1.6%. Under these conditions the optimum damping frequency was .43 Hz with 7.6% internal damping.

As in the testing of pendulum model A, a .00635 M diameter threaded rod with root radius of .00254 M was chosen for the damping. Testing conducted to confirm the rod's properties revealed 1.12% internal damping at .669 M effective rod length. This rod length produced oscillations with periods of approximately 2.29 seconds. The period although not ideal, was considered within the range of error found in producing the .432 Hz optimum damping frequency with the rod. The increased internal damping most likely resulted from variation during manufacture.

The system parameters have been compiled in Table 5.2. The system parameters for Pendulum Model A and it's damping rod have also been included for comparison.

Pendulum Model B			Cantilevered Damping Rod For B		
Payload Mass	=	122 Kg	Damping Mass	=	2.0 Kg
Frequency	=	.439 Hz	Optimum Frequency	=	.43 Hz
Internal Damping	=	0.02 %	Optimum Internal Damping	=	7.6 %
Release Displacement	=	7.24 Degrees	Actual Internal Damping	=	1.12 %

Pendulum Model A			Cantilevered Damping Rod For A		
Payload Mass	=	122 Kg	Damping Mass	=	2.0 Kg
Frequency	=	.44 Hz	Optimum Frequency	=	.43 Hz
Internal Damping	=	1.32 %	Optimum Internal Damping	=	7.6 %
Release Displacement	=	7.24 Degrees	Actual Internal Damping	=	0.04 %

Table 5.2 Pendulum Model B and Damper

Although the rod's internal damping was not raised to the optimum value of 7.6%, pendulum model B's internal damping was reduced to .02%. If the analysis of pendulum model A's testing was correct, the new testing would achieve the predicted tuned mass damping results. In this case the damper would absorb and dissipate the pendulum's energy in a cyclic manner similar to Graph 2.1. Because the damper did not have the optimized internal damping, it was predicted that some energy would be returned to the pendulum. However, over the sampling period, tuned mass damping should have been noted. This would be depicted in both graphs of sensor output and graphs of pendulum motion. Likewise, graphs of Fourier transform tests were expected to indicate tuned mass damping.

5.7 Pendulum Model B Testing

The testing of pendulum model B with the tuned cantilevered rod followed the same testing procedure as pendulum model A. This was considered important to confirm the value of altering the pendulum's internal damping.

From the initial test it was apparent that a large energy transfer had occurred. Unlike previous testing, the cantilevered rod damper exhibited large displacement within the first half of the release cycle. The damping displacement was so large, that the cantilevered rod failed by bending to an unusable form. The rod visibly appeared to be bent close to 90 Degrees at mid-length. Further tests were conducted at decreased release displacements to reduce the amount of energy within the pendulum and damper system. Similar rod failures occurred over the next 5 tests. Upon decreasing the initial displacement release to .04 M or 1.8 Degrees, a complete testing cycle was achieved.

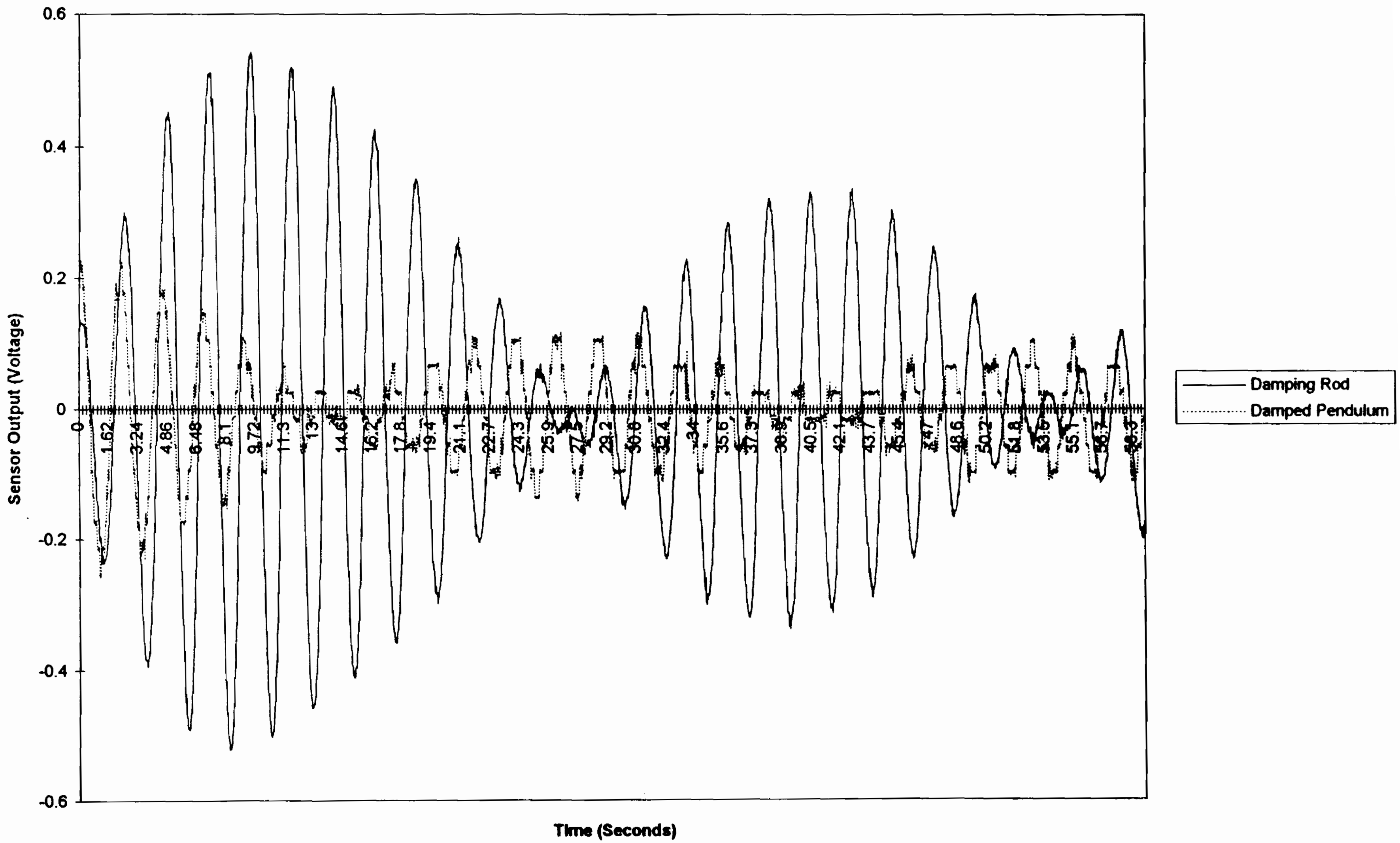
Tests were conducted to evaluate the damper's effect upon pendulum model B from the new initial displacement. Data was taken with varied damper frequencies above and below the optimized value. The rod frequencies were once again defined by increasing and decreasing the effective rod length.

Graphs 5.8, 5.9, and 5.10 depict the sensor output for testing with pendulum model B. It is evident that a cyclic energy transfer is occurring between the damper and pendulum. As pendulum oscillation energy is transferred to the damper, the pendulum is brought to rest. The increased damper oscillation energy amplifies the rod's motion as sampled by the accelerometer. As predicted, the damper's low rate of internal damping prevents optimum energy control and dissipation. It is apparent however, that the pendulum is brought to rest within 5 cycles for the first energy transfer. Likewise the oscillations decrease over the 60 second sampling period as a result of the damping.

Graphs 5.11, 5.12, and 5.13 depict pendulum B's displacements as compared to an undamped test. With the optimized rod frequency, Graph 5.11, the damper's displacements were reduced from .04 M to approximately .005 M within 5 cycles or 12 seconds. At such a small displacement sampling noise prevents accurate interpretation of the pendulum's true motion. However, it is clear that an 85% or greater displacement reduction has occurred. The .005 M displacements would correspond to .22 Degrees of swing from center. If similar results were achieved on a 100 M construction crane, the payload displacement would be reduced from a maximum of 3.5 Degrees or 6 M to less than half a meter from center.

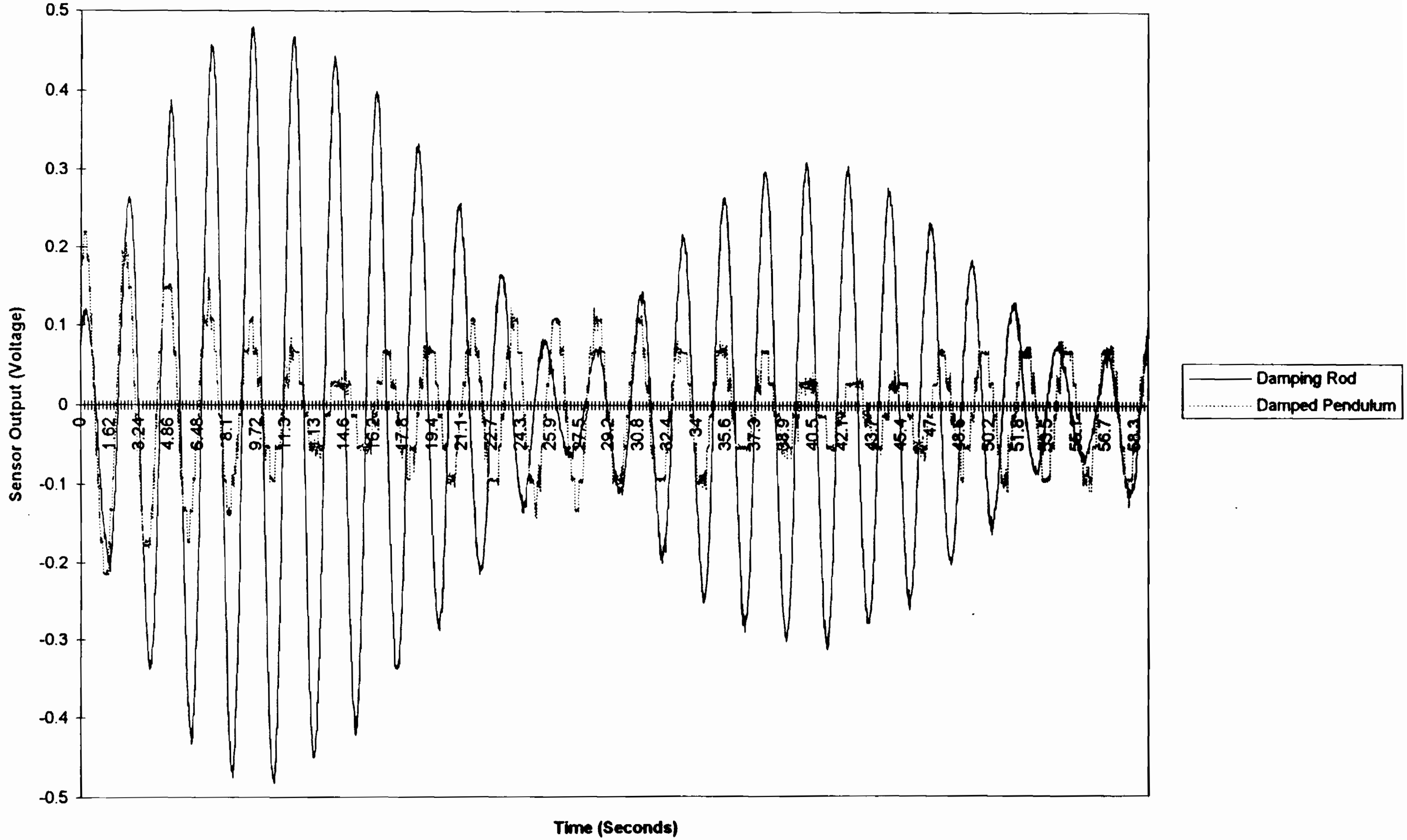
Comparing the graphs further gives an indication of the importance of damper optimization. Graphs 5.9, 5.10, 5.12, and 5.13 provide results from damping tests with frequencies above and below the optimum value. It is clear that the optimized damper achieved the greatest oscillation reduction. It is also evident that the time required to achieve the greatest damping changed as a result of damper frequency adjustment. Damper tests with decreased frequencies achieved the damping in the shortest time period. However, the optimized frequency developed a greater oscillation reduction in only 1 more second. This would confirm the use of damper optimization based upon mass ratio's.

1.67% Mass Ratio Optimized Rod Frequency



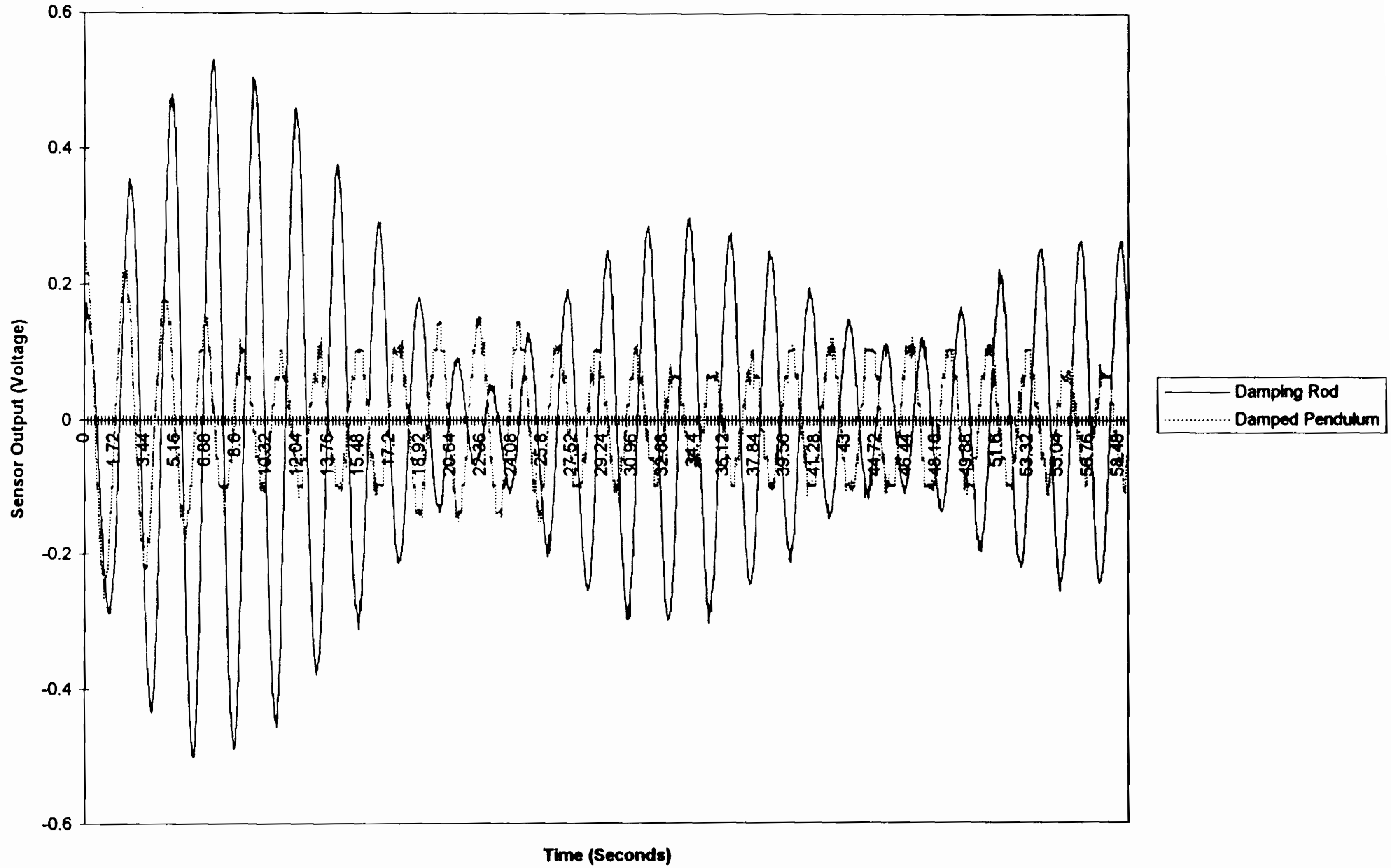
Graph 5.8 Pendulum Model B Testing

1.67% Mass Ratio Higher Rod Frequency



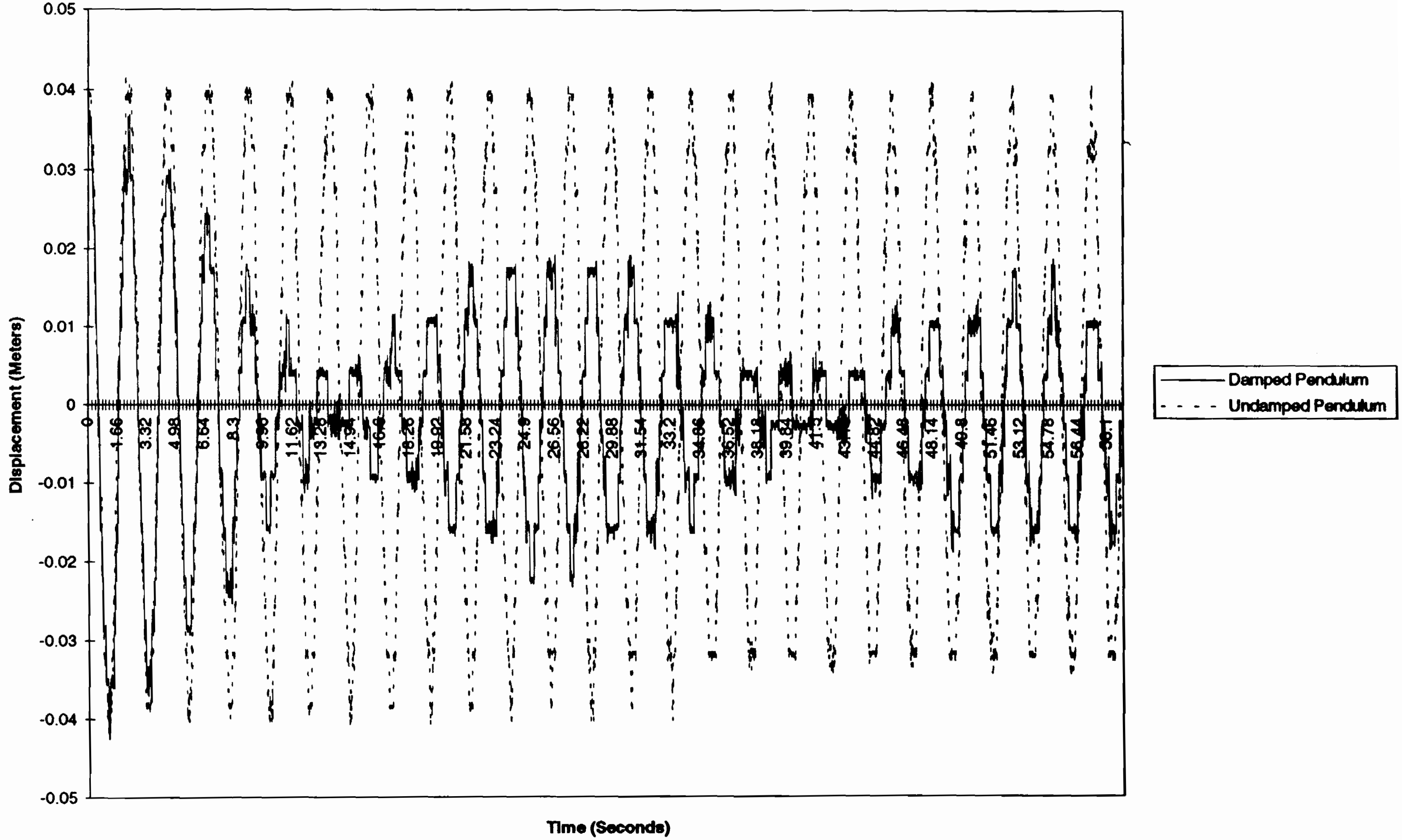
Graph 5.9 Pendulum Model B Testing

1.67% Mass Ratio Lower Rod Frequency



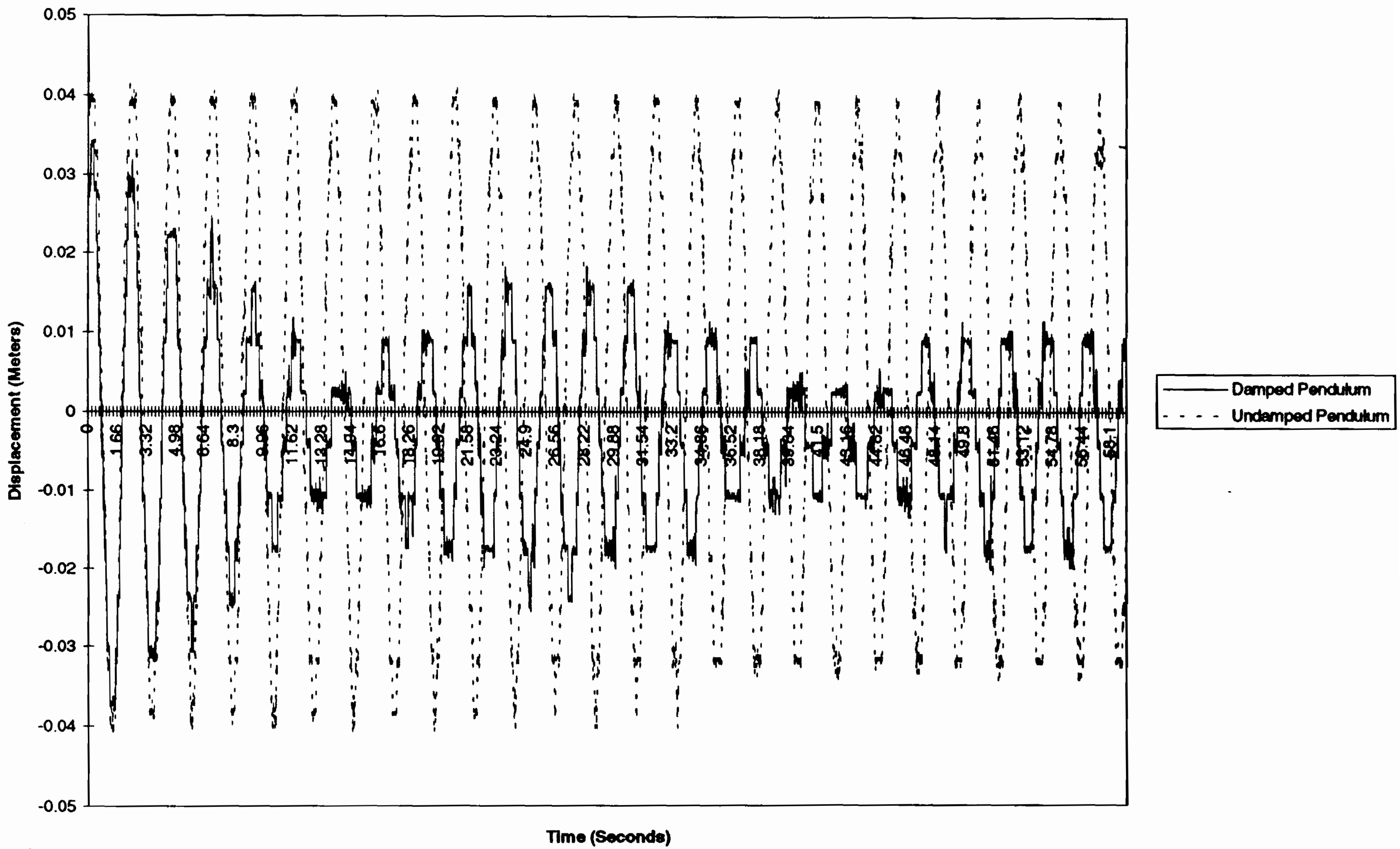
Graph 5.10 Pendulum Model B Testing

1.67% Mass Ratio Optimized Rod Frequency



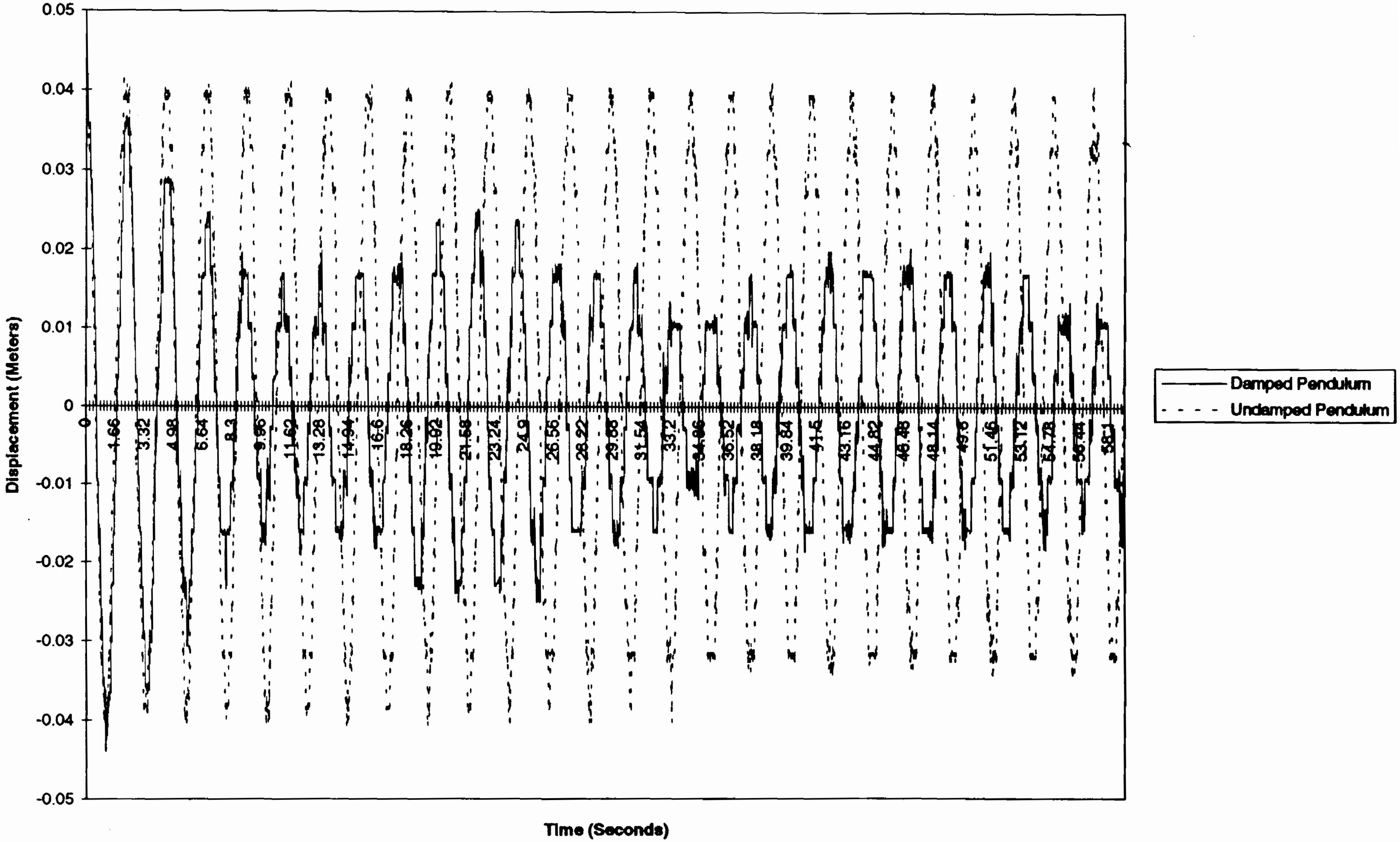
Graph 5.11 Pendulum Model B Testing

1.67% Mass Ratio Higher Rod Frequency



Graph 5.12 Pendulum Model B Testing

1.67% Mass Ratio Lower Rod Frequency

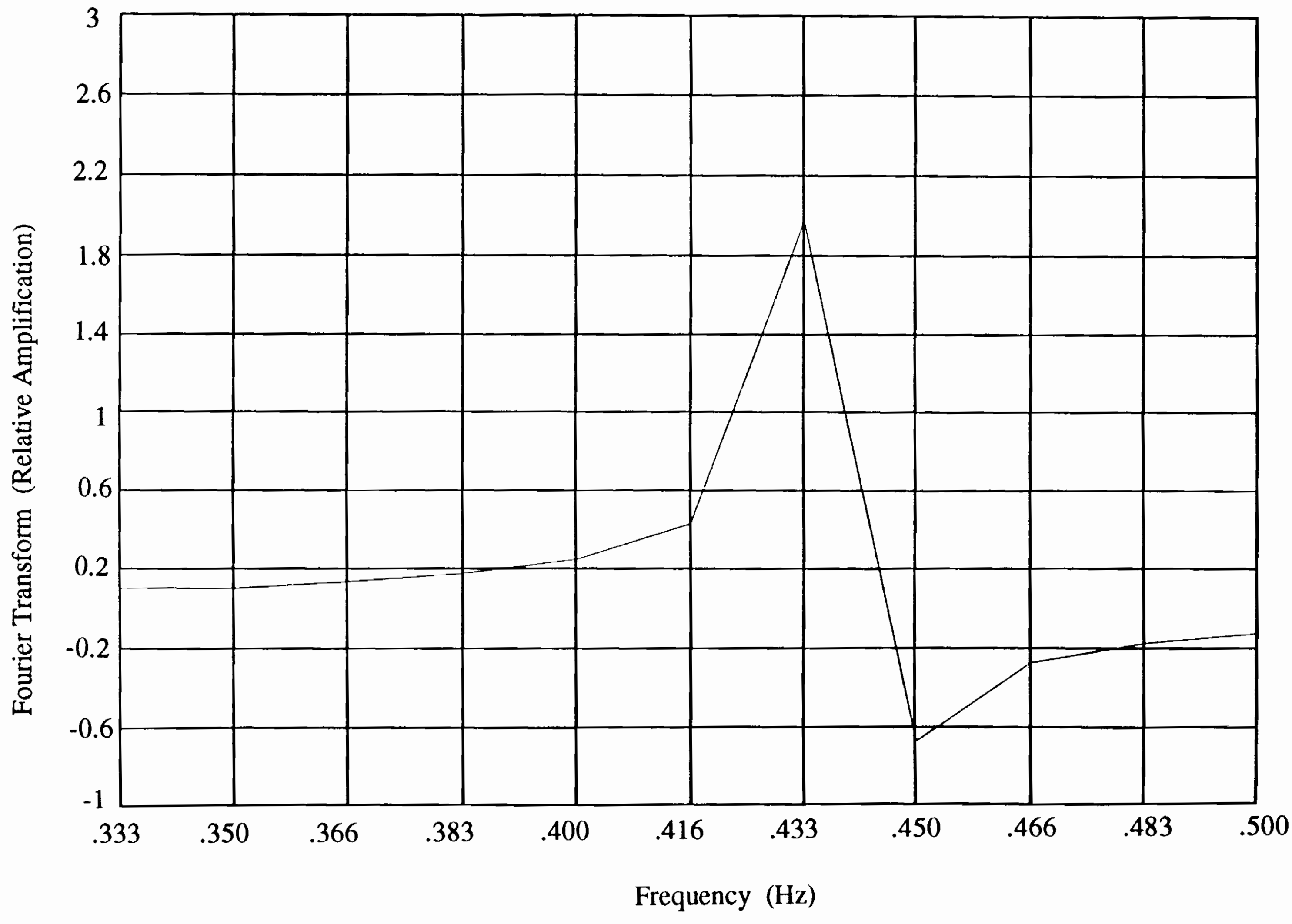


Graph 5.13 Pendulum Model B Testing

As a second analysis, Fourier transforms were calculated for the sampled data. Graph 5.14 represents the single frequency amplification peak corresponding to the oscillations of an undamped pendulum model B. Graphs 5.15, 5.16, and 5.17 depict the Fourier transforms with the attached damper. It is clear that two oscillation peaks of decreased amplitude were achieved in testing with the operational damper. This is consistent with the theory of tuned mass damping. Further examination reveals the value of frequency optimization. In Graph 5.16, the optimized damper achieves frequency peaks of relatively the same amplitude. A more exact rod tuning would be expected to reach equal amplitude frequency peaks.

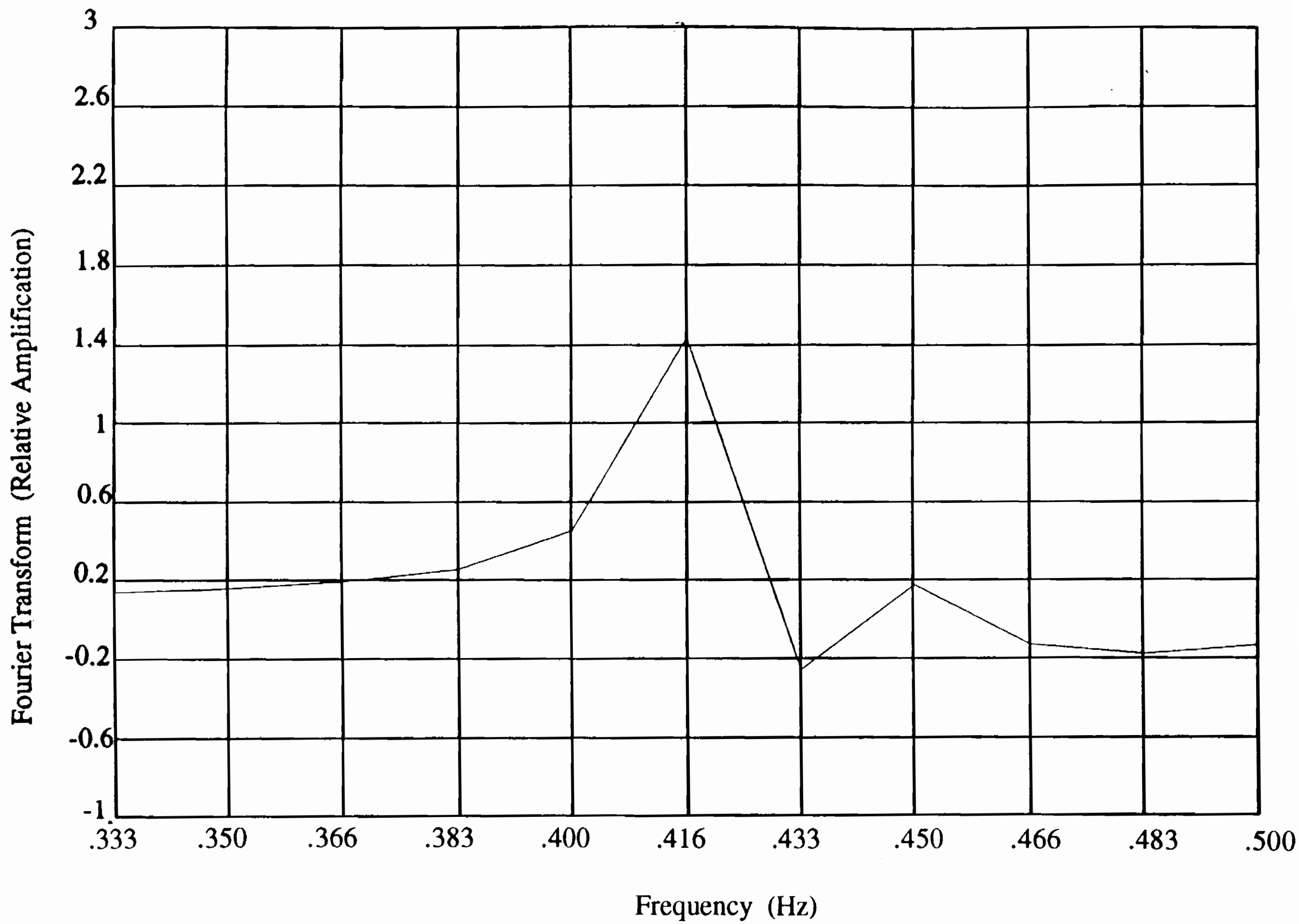
The value of tuning the rate of internal damping also becomes apparent in evaluating the graphs. As Brock's research suggested (Smith, 1988), tuned internal damping will create flat frequency peaks. In Graph 5.16, the twin peaks do not appear to be flat in representation. While the resolution of the graph has limited the curves smoothness the large valley between the peaks indicates the lack of tuned internal damping.

Undamped Fourier Transform

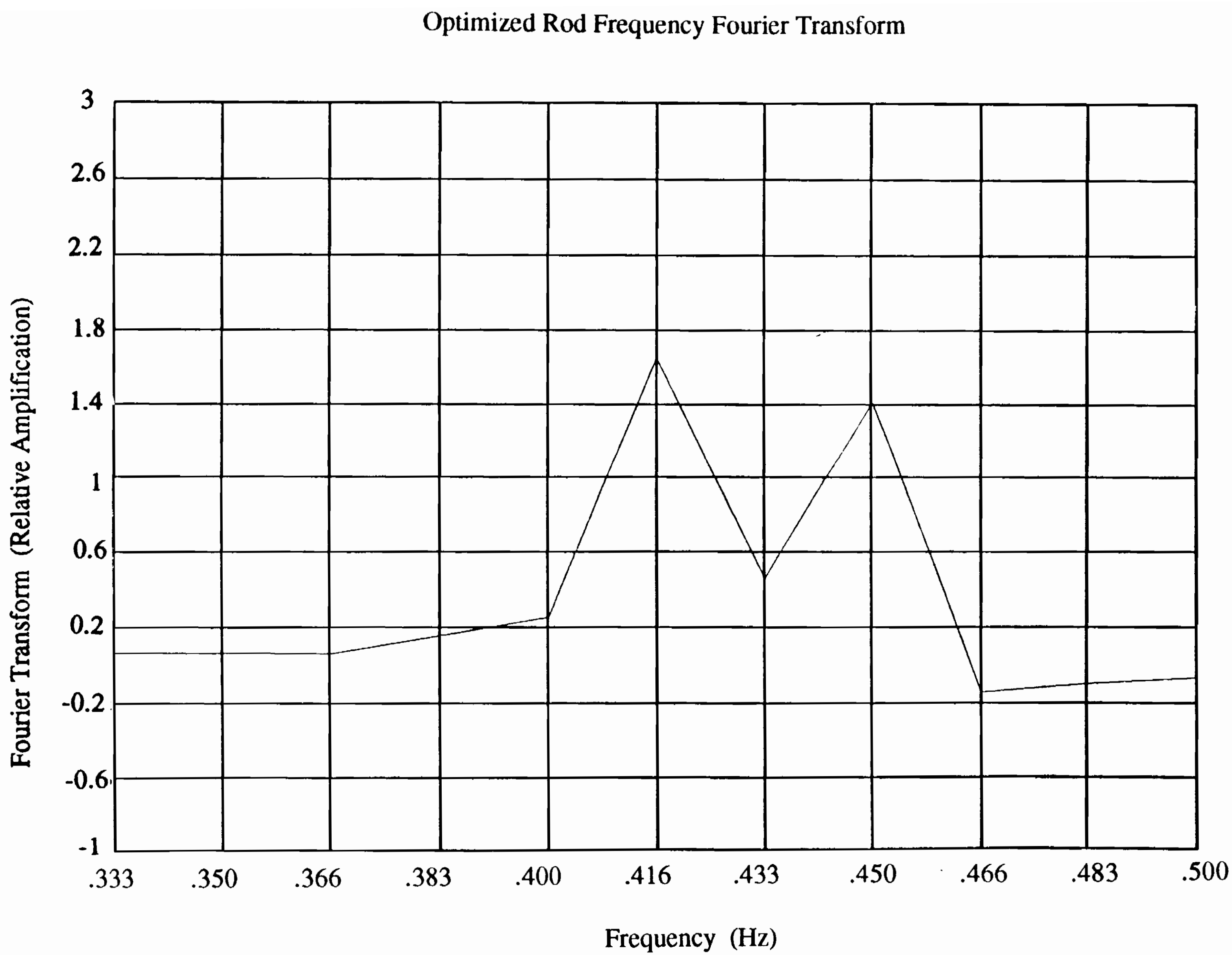


Graph 5.14 Pendulum Model B Testing

Higher Rod Frequency Fourier Transform

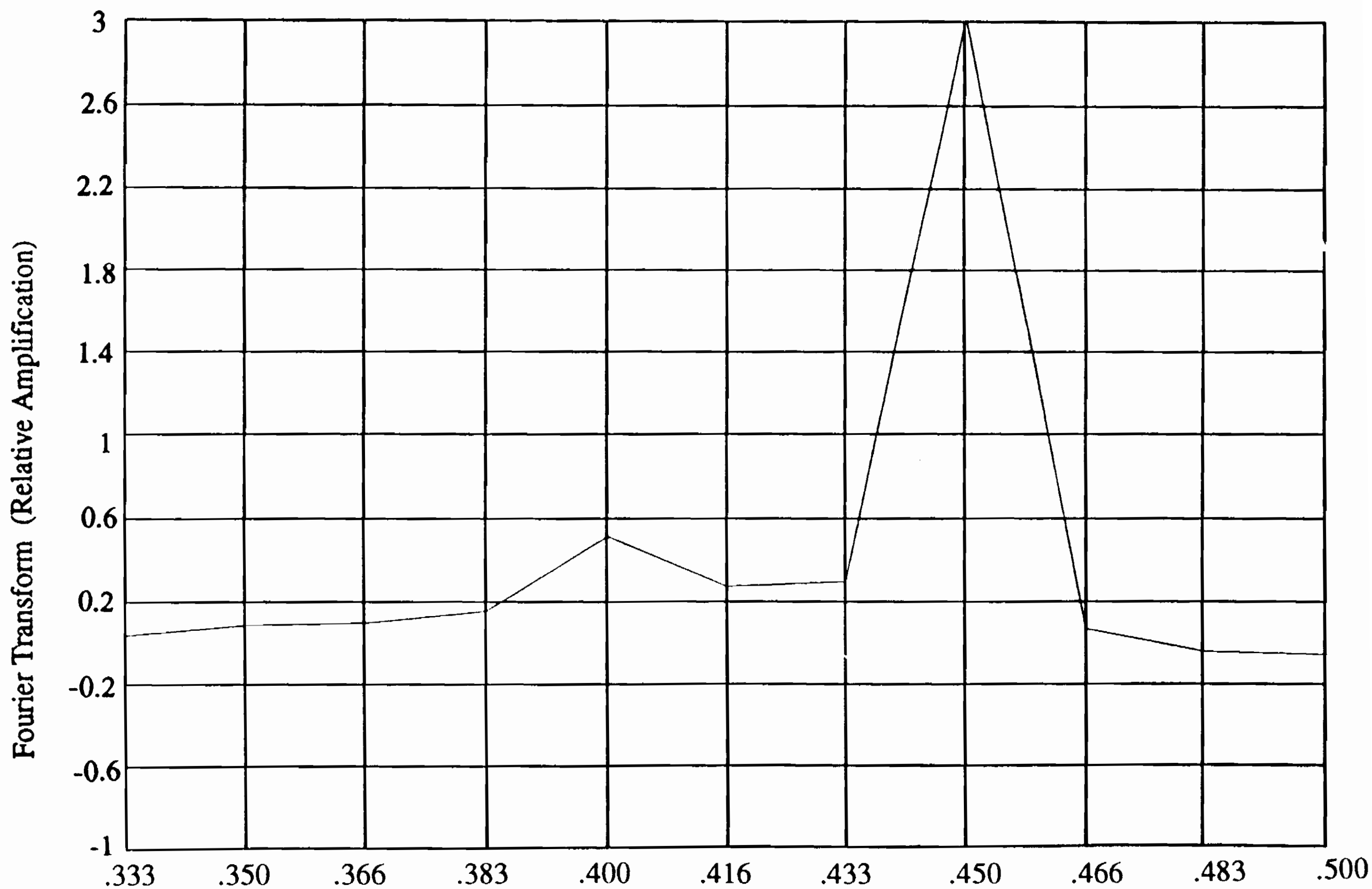


Graph 5.15 Pendulum Model B Testing



Graph 5.16 Pendulum Model B Testing

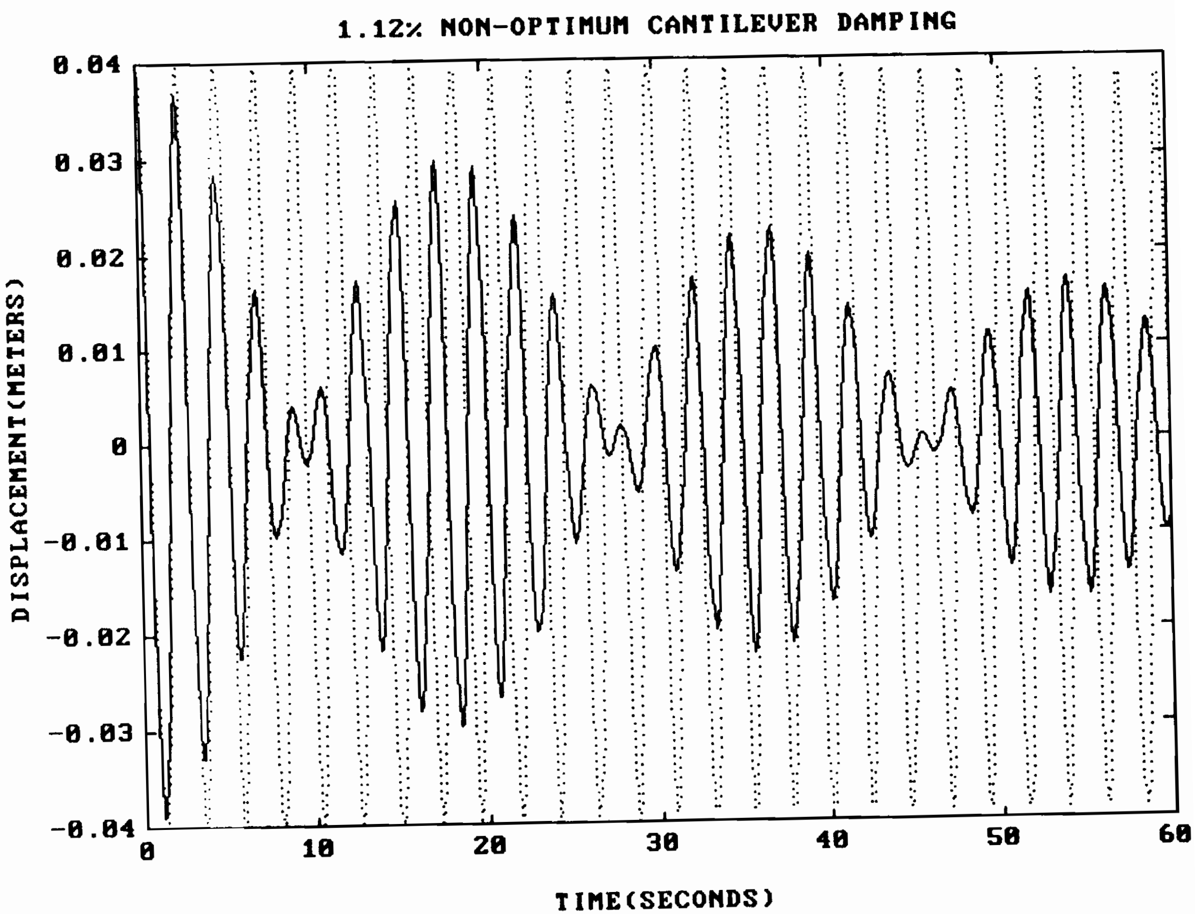
Lower Rod Frequency Fourier Transform



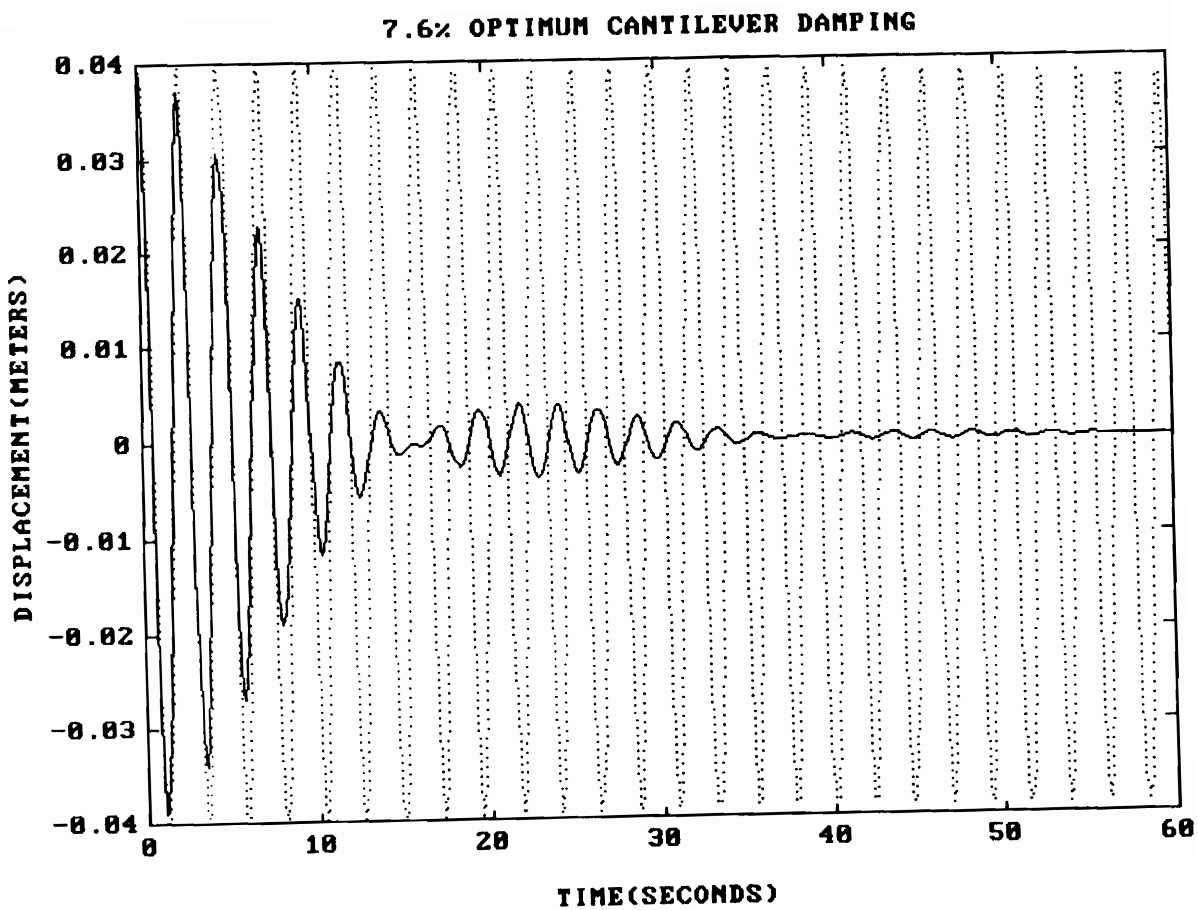
Frequency (Hz)

Graph 5.17 Pendulum Model B Testing

To confirm the testing results, computer simulation was conducted. The Matlab code included in appendix A was adjusted to reflect the system parameters present during physical testing. Graph 5.18 represents the simulation of cantilevered rod damping with exactly tuned frequency and non-optimized internal damping. The results are very similar to those of pendulum model B as shown in Graph 5.11. A second simulation with the addition of optimized internal damping produced Graph 5.19. The optimized internal damping significantly reduced the energy transfer from the damper back to the pendulum by controlling and dissipating the energy. This appears to confirm the importance of optimized internal damping as suggested by the researcher.



Graph 5.18 Non-optimum Damping Simulation



Graph 5.19 Optimum Damping Simulation

CHAPTER 6

CONCEPTUAL CRANE MODEL

The success of the tuned cantilevered rod in testing suggests that a similar system could be viable for application on construction cranes. This system would follow much of the design procedure used in the creation of the tuned cantilevered rod damper. Key topics include the damper's ability to reach the frequencies required for construction crane damping. The system must also be capable of adjusting to new frequencies during operation. The ability to adjust the rate of internal damping would also be required. Other considerations include the damper's capacity to function under most weather and site conditions without failure. Low maintenance and ease of retro-fitting are additional topics.

6.1 Cantilevered Rod System

Much of the cantilevered rod's function met the design requirements. The cantilevered rod was inexpensive and readily available. The design required few moving parts and low maintenance. Various frequencies could be achieved by simply changing the effective rod length. In addition, the system was light and easily attached.

Throughout the testing procedure though, the cantilevered rod's specific application for construction crane damping became increasing questionable. Although the design was viable for testing purposes and did achieve the desired results, several important problems arose. To achieve the desired model frequencies it was necessary to increase the rod length to a greater size than anticipated. This resulted from both the light damping load required and the effects of the cyclic loading. Attempts to use smaller radii rod's to decrease rod lengths resulted in rod failures. It is possible that the frequency equation was not applicable at small rod dimensions when considering the critical buckling load.

The cantilevered rod had a second design failure. Tuned mass damping resulted in greater rod fatigue than predicted. The energy transfer from the pendulum to the damper resulted in very large oscillation displacements. This would prove to be an interference in crane operation. System failure is highly probable under large payload oscillations as observed on pendulum model B.

6.2 Alternative Solutions

Although spring systems are typically incapable of achieving slow rates of oscillation, a spring damping system has been designed for tuned mass damping. The lever and pendulum mechanism was developed by Fujinami et al. (Fujinami, et al., 1990) for application on buildings and towers. Much of the design theory presented within this thesis was followed in the lever and pendulum design. Long periods were required as well as ease of operation. Design constraints like reduced oscillation amplitudes and the corresponding reduction in fatigue resulted from the limited building area available for damper operation. The lever and pendulum tuned mass damper is represented in plan view in Figure 6.1.

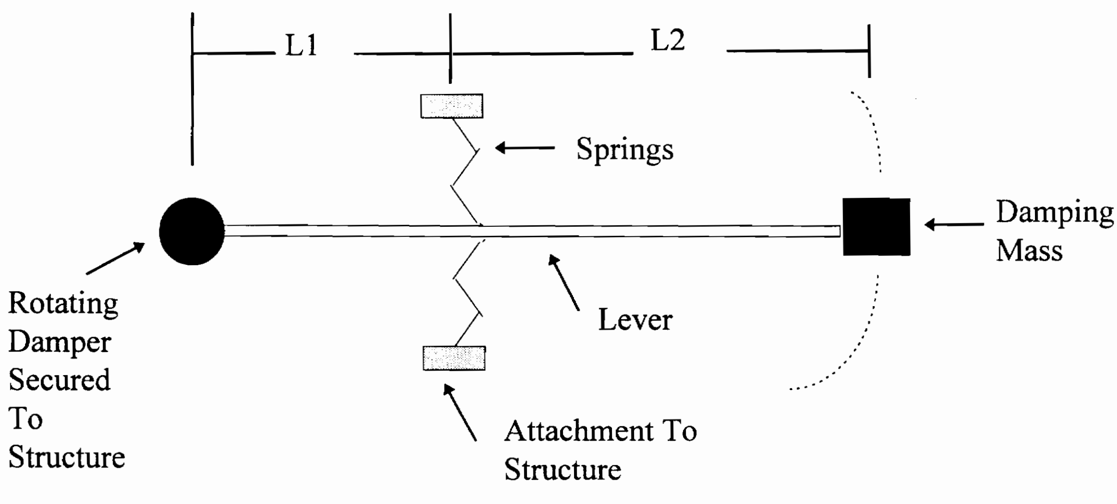


Figure 6.1 Lever and Pendulum Tuned Mass Damper

The system oscillation frequency is based upon the lever ratio, L2 / L1 (Fujinami, et al., 1990).

Equation 9 $$f = \frac{1}{2\pi}\sqrt{K / M\left\{1 + \mu(1+\lambda)^2\right\}}$$ Lever Frequency

Where K is spring stiffness, M is absorber mass, μ is pendulum mass / damper mass, and λ is lever ratio L2 / L1.

Following the design principles developed in the paper, the lever and pendulum system appears to be capable of operating on a construction crane. Operation occurs in the horizontal X - Y plane. This reduces the damper's height and corresponding affect upon crane performance. Frequency adjustment is achieved by changing the lever ratio between L1 and L2. This also changes the amplitude of oscillation and fatigue. Using the lever ratio allows the designer to incorporate stronger springs with increased fatigue resistance. Adjustment of the system's internal damping is controlled in the rotating damper which forms the lever's base.

In theory the design could be incorporated in a manner similar to the cantilevered rod. A series of 2 or 3 lever and pendulum dampers would be positioned to control oscillation throughout the two dimensional plane as depicted in Figure 6.2. The systems would be stacked above the payload on a spreader bar hung directly from the crane cable.

Frequency adjustment would involve motor controlled movement of the springs to alter the lever ratio. Adjustments could also be provided in the damper to achieve optimized rates of internal damping.

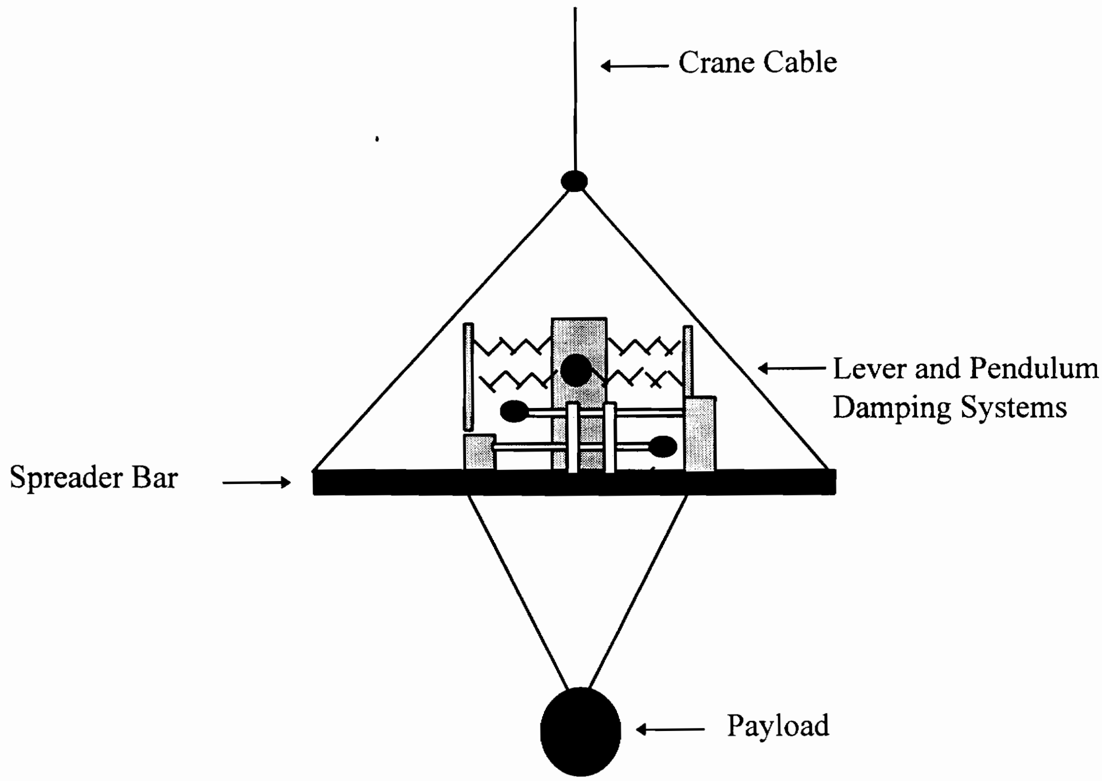


Figure 6.2 Lever and Pendulum Dampers On Crane

The control system would conceptually involve the sampling of payload motion with 3 dimensional accelerometers powered by battery packs. A shackle tension device would be incorporated to monitor payload mass. The measurements would be transferred to a central computer control using radio frequency modems. The computer control would compute optimized frequencies and rates of internal damping based upon the accelerometer and shackle tension data.

CHAPTER 7

SUMMARY AND CONCLUSIONS

7.1 Summary

The tuned mass damping concept has been shown effective for damping crane oscillations modeled as pendulum type motion. This was confirmed in both computer simulation and physical testing.

As suggested in testing, a frequency optimized, tuned mass damper is capable of reducing payload oscillations from an initial displacement of 1.8 Degrees to .22 Degrees in 12 seconds or 5 cycles. This would require a damper to payload mass ratio between 1 and 2%. The same damper with the addition of tuned internal damping has been predicted through simulation to be capable of achieving better results.

The use of cantilevered rod tuned mass damping devices as presented in this thesis is inappropriate for application on construction cranes. Testing suggested that damper size limitations would be exceeded. The ability to reach long periods of oscillation was questioned due to buckling. In addition, damper mass oscillation on cantilevered rods would interfere in crane operation. Failure under cyclic loading extremes caused by uncharacteristically large pendulations was also confirmed.

Tuned cantilevered rods were shown to be effective as tuned mass damping devices. The system limitations are specific to the frequency and displacement extremes present in cranes. Applications such as bridges and buildings oscillate at significantly higher frequencies than cranes. Likewise, the amplitude of oscillation on bridges and buildings is typically smaller than that of cranes. Under these conditions a tuned cantilevered rod could be used effectively for damping as suggested by the research conducted in this thesis.

7.2 Recommended Uses and Significance

The testing and conceptual modeling presented in this thesis is recommended for use as a building block in future research and design. The concept of tuned mass damping has been characteristically applied to bridge and floor damping systems. The ideas presented here represent an extension of tuned mass damping theory to the topic of construction crane damping.

This thesis is also recommended as a conceptual design analysis of semi-passive damping applications on construction cranes. To date, a commercial damping system of this nature has not been commonly applied in operation. It is suggested that the design of damping systems for construction crane payload oscillations would be advantageous. The reduction of construction costs and schedule delays in addition to site injury are considered motivational issues.

7.3 Future Extension

The models and experiments presented here are considered an initial investigation of tuned mass damping for application on construction cranes. Future research extension can be broken into two topic areas, crane experimentation and tuned mass damping experimentation.

Crane experimentation would be vital to the success of a damping system in commercial application. Few studies have effectively analyzed the effects of payload oscillation on productivity in construction. While cranes have been cited as critical components in construction, their function and shortcomings are relatively unexplored to date. Topics for evaluation would include.

1. Time and efficiency studies of commercial crane operations.
2. Direct and indirect operation costs resulting specifically from payload oscillation.
3. Rate of internal damping present in construction cranes.

The subject of tuned mass damping could benefit greatly from future experimentation. Although the concept of tuned mass damping has been well documented, commercial application is still not commonly practiced. Motion evaluation using 3 dimensional accelerometers could be valuable in further defining the principles of the concept. Other topics would include the following.

1. Damper design for structures with low frequencies, like cranes.
2. Control theory for expansion to semi-passive damping.
3. Design intended to replace active damping systems.

7.4 Conclusions

The increasing reliance upon cranes in all facets of commercial construction has deemed it necessary to investigate control techniques. Current designs and operational methods have failed to reduce payload oscillations to a controllable state. Damping systems designed for payload oscillations on overhead cranes has been proven effective. Similar systems could be designed for construction cranes to reduce construction costs and improve safety.

The concept of tuned mass damping for construction payload oscillations is considered viable. The system's passive operation is beneficial to the creation of a robust, semi-passive damper. This stems from the reduction of complex parts and the reliance upon a natural reaction between structures.

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APPENDIX A

MATLAB CODE FOR SIMULATIONS

```
% thsim.m crane problem; May 31, 1995 (Jean-Guy Beliveau)
format compact
diary thsim.txt
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
n=2                 %number of degrees of freedom
n2=n*2;             %order of matrices used in matrix exponential solution
zeta1=.0002         %damping ratio of bucket pendulum
zeta2=.0112         %damping ratio of damping system
m1=122              %mass of bucket(kg)
G=9.81              %gravity acceleration (m/s/s)
l1=1.3              %length of bucket pendulum(meters)
x1o=.04             %initial displacement of bucket(m)
v1o=0               %initial horizontal velocity of bucket(m/s)
x2o=.04
v2o=0
m2=2                %tuned mass(kg)
w2=m2*G             %weight of tuned mass
k1=m1*G/l1          %stiffness of bucket pendulum
gama=12/pi/pi       %parameter for cantilever change in potential energy
to=0;tmax=60;NT=501;t=linspace(to,tmax,NT)';
masratio=m2/m1
om1=sqrt(k1/m1)
om2=om1/(1+masratio)
%optimum stiffness
k2=m2*om2^2                                               %(1)
%second pendulum properties
%l2=50                %length of damped mass of double pendulum
%k2=w2/l2             %equivalent stiffness of tuned pendulum    %(2)
%cantilever using smith formula
%e=205e9
%r=.00254
%ei=e*pi*r^4/4
%h=.669
%k2=pi^2*(ei*pi^2/32/h^3-m2*G/8/h)                        %(3)
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
```

```
om1=sqrt(k1/m1)             %resonant frequency of bucket pendulum
om2=sqrt(k2/m2)             %resonant frequency of tuned mass damper
M=[m1 0;0 m2]               %mass matrix
K=[k1+k2 -k2;-k2 k2]        %stiffness matrix
[u,l]=eig(K,M);             %eigenvalues and eigenvectorss
[o2,rank]=sort(diag(l));    %square of resonant frequencies increasing order
uu=u(:,rank);               %ranked eigenvectors
ome=diag(sqrt(o2));         %ranked frequencies in (radians/second)
norm=uu'*M*uu;              %generalized mass matrix
for J=1:n
div=uu(n,J);
for I=1:n
u(I,J)=uu(I,J)/div;phi(I,J)=uu(I,J)/sqrt(norm(J,J));
end
end
u
phi
c1=2*zeta1*sqrt(k1*m1)        %damping for given modal damping ratio
c2=2*zeta2*sqrt(k2*m2)        %damping for given modal damping ratio
C=[c1+c2 -c2;-c2 c2]          %damping matrix
%3)find time history response for initial displacement of 1
%using matrix exponential and single degree of freedom
yo=[x1o;x2o;v1o;v2o]
%find state space eigenvectors for damped structure
A=[zeros(n) -K;-K -C];B=[-K zeros(n);zeros(n) M];
[v,ll]=eig(A,B);[lamb,rank]=sort(diag(ll));vv=v(:,rank);
lambt=lamb'
for J=1:n2
div=vv(n,J);
for I=1:n2
v(I,J)=vv(I,J)/div;
end
end
v
vi=inv(v);
```

```
%plot time history response
nu=-zeta1*om1
mu=om1*sqrt(1-zeta1^2)
voxo=(v1o+nu*x1o)/mu;
for I=1:NT
y=v*expm(diag(lamb*t(I)))*vi*yo;nut=nu*t(I);mut=mu*t(I);
x0(I)=exp(nut)*(x1o*cos(mut)+voxo*sin(mut));
x1(I)=y(1);x2(I)=y(2);x21(I)=y(2)-y(1);
end
diary off
disp('plot(t,x0,t,x1,t,x2,t,x21)')
```

VITA

The author received a Bachelor of Science degree from the Charles Edward Via, Jr. Department of Civil Engineering at Virginia Polytechnic Institute and State University. Upon graduating in December of 1993, he began work to complete a degree of Master of Science in Civil Engineering within the same department. This thesis is the result of research conducted throughout this period. Upon receiving his Master of Science, the author began work within private industry.